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Demonstration Truck Program 4. Noise Reduction Technology and Costs for an International Harvester F-4370 Heavy-Duty Diesel Truck

Bolt Beranek and Newman, Inc.
Cambridge, MA

Prepared for

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Noise Reduction Technology and Costs for an International Harvester F-4370 Heavy-Duty Diesel Truck

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Demonstration Truck Program

4

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This is one in a series of seven technical reports and a program summary prepared for the Environmental Protection Agency's Demonstration Truck Program. The reports in this series are listed below.

Report Number	Title	Date
1.	Program Summary: Truck Noise Reduction (BBN Report No. 4839).	December 1981
2.	Noise Reduction Technology and Costs for a Ford CLT 9000 Heavy-Duty Diesel Truck (BBN Report No. 4379).	October 1981
3.	Noise Reduction Technology and Costs for a General Motors Brigadier Heavy-Duty Diesel Truck (BBN Report No. 4507).	October 1981
4.	Noise Reduction Technology and Costs for an International Harvester F-4370 Heavy-Duty Diesel Truck (BBN Report No. 4667).	October 1981
5.	Noise Reduction Technology and Costs for a Mack R686 Heavy-Duty Diesel Truck (BBN Report No. 4795).	December 1981
6.	Field Test of a Quieted Ford CLT 9000 Heavy-Duty Diesel Truck (BBN Report No. 4700).	October 1981
7.	Field Test of a Quieted General Motors Brigadier Heavy-Duty Diesel Truck (BBN Report No. 4796).	December 1981
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This report discusses the technology and costs required to reduce the noise of an International Harvester F-4370 heavy-duty diesel truck from 81.1 to 72.2 dBA. The noise control treatment consists primarily of a dual exhaust silencing system and a partial enclosure for the engine and transmission. The noise treatment increases the vehicle weight by 332 lb and estimated vehicle price by $1302. Wind tunnel tests on the completed truck show that temperatures of engine coolant and oil remain within generally acceptable limits.

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NOISE REDUCTION TECHNOLOGY AND COSTS FOR AN INTERNATIONAL HARVESTER F-4370 HEAVY-DUTY DIESEL TRUCK

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PREFACE

This report deals with the technology and costs of treatments developed and implemented by Bolt Beranek and Newman Inc. (BBN) to reduce the noise level of an International Harvester F-4370, one of the heavy-duty diesel trucks in the Environmental Protection Agency's Demonstration Truck Program. This program, begun in 1979, included four heavy-duty diesel trucks, each with a different engine. The original program plan called for each vehicle to receive noise reduction treatments and then to enter fleet service for a year of field testing. Each of the four vehicles successfully completed the noise reduction part of the program. The duration of the program was shortened from the original plan; thus only two of the vehicles completed an entire year of field testing. The third truck was in supervised field service for five months, and the fourth truck did not enter fleet service.

The focus of the Demonstration Truck Program was on the technology of treating the vehicles, rather than components such as engines or tires. The EPA conducted parallel programs on diesel engine and tire noise control; these other programs were to be integrated with the truck program. Accordingly, BBN's treatment has been primarily to add mufflers for exhaust noise control, enclosures for engine and transmission airborne sound, and vibration isolators for engine structureborne sound where required.

Seven final reports and a program summary were prepared by BBN for the Demonstration Truck Program. Their titles are listed on the inside cover of this report. The reports appeared in draft version beginning in early 1980 and extending through 1981. The final version of each report was prepared in late 1981. Each of the reports is intended to be internally complete; therefore, some redundancy occurs among the four technology and cost reports.

a reader who has already read one technology and cost report will find that he can pass over the nearly identical introduction and test requirements sections (Sec. 1 and Appendix A) and focus on the remaining sections that contain unique technical material.

The authors are grateful to the many governmental and industrial organizations and personnel who have contributed to the development of the noise treatment for this truck. The program has been sponsored by the Environmental Protection Agency's Office of Noise Abatement and Control. The International Harvester Company provided technical information on the truck. The Cummins Engine Company performed cooling tests at its facility in Columbus, Indiana. The Donaldson Company supplied the exhaust silencing system, and Tech Weld fabricated many of the engine/transmission enclosure components. Noise testing was done at Hanscom Field with the cooperation of the Charles Stark Draper Laboratories and the Massachusetts Port Authority.

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1. INTRODUCTION

The primary objective of the project described in this report has been to reduce the noise level of an International Harvester F-4370 Transtar heavy-duty diesel truck from 81.1 to 72 dBA at 50 ft. This target level, established by EPA, is lower than the level of any heavy diesel truck in current production, and has been reached on only three other roadworthy U.S. trucks in recent history [1-4]. An additional objective, also established by EPA, is to ensure that cab noise levels do not exceed 78 dBA. This level corresponds to a proposed interior bus noise level of 80 dBA [5], less 2 dBA to account for manufac-turing tolerances.

To be acceptable, the noise treatment must allow the truck to function in a normal manner. Accordingly, the treatments must be durable, interfere as little as possible with maintenance activities, add as little weight as possible, permit continued adequate component cooling, and have minimal impact on engine efficiency. All of these factors may be characterized in terms of equipment and operating costs. Projections of initial equipment costs will be treated here; operating costs will be determined during the course of a subsequent in-service evaluation.

The technical approach to the development of noise treatment for the IH F-4370 has involved four major phases:

I. Baseline noise testing
II. Development of noise control treatments
III. Final noise and cooling tests
IV. Equipment cost estimation.

In the first phase, the untreated vehicle is noise-tested at EPA's Noise Enforcement Facility at Sandusky, Ohio. The vehicle is then delivered to BBN's facility in Cambridge, Massachusetts, where we conduct exterior noise measurements. Diagnostic tests are also performed to determine contributions from major noise sources (intake, exhaust, tires, engine, and transmission). Quantitative goals for each source are established and compared to the actual contributions. The differences then become the noise reduction objectives that must be achieved by each treatment for the entire vehicle to reach the 72-dBA level.

In the second phase, we develop the noise treatment, which consists primarily of an exhaust silencing system and an engine/transmission enclosure. The exhaust system is first laboratory-tested to ensure that it meets our goals and then installed on the truck. A mockup enclosure, built of 1/4-in. Masonite and fiberglass, is tailored to the vehicle. These inexpensive and easy-to-form materials are used because of the cut-and-fit approach that is needed to conform to the complex geometry associated with the truck and its many components.

After a suitable mockup enclosure is developed and tests are performed to indicate that goals have been met, the enclosure is fabricated from metal and sound-absorptive materials, and installed in a nearly final form. In this phase, some refinements are implemented to tune the system acoustically, thereby bringing the vehicle into closer compliance with the goals.

In Phase III, the truck undergoes final noise testing and wind tunnel testing to ensure that cooling requirements are met. In addition, the vehicle and available data are reviewed by EPA, the vehicle manufacturer, and the fleet operator to verify, insofar as practicable, that the vehicle is ready for service.* The technical development is then complete and the truck enters fleet service.

While costs are taken into account qualitatively in the numerous decisions made throughout the program, a formal cost assessment is deferred until the vehicle is complete. At this point (Phase IV), a formal detailed equipment cost analysis is performed.

Section 2 of this report describes the baseline truck and the noise source levels associated with its major components. Section 3 presents a discussion of the noise treatment. The final interior and exterior test data are summarized in Sec. 4. The performance of the engine cooling system is evaluated in Sec. 5, and the incremental costs and purchase prices associated with the noise treatment are estimated in Sec. 6. Noise test procedures are briefly summarized in Appendix A. Appendices B and C describe procedures for the estimation of source contributions and structureborne noise.

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*Members of the reviewing organizations apply engineering judgment but do not conduct detailed engineering analyses or tests.

2. BASELINE TRUCK CONFIGURATION AND NOISE LEVELS

2.1 Truck Description

The baseline truck, as received by BBN at the beginning of the noise treatment project, is illustrated in Fig. 1. It is an IH Model F-4370 long conventional 6 × 4 tractor with a 162-in. wheel base. The cab has a 117-in. length (BBC). Fully fueled, but without a driver, the tractor weighs 14,048 lb; it has a gross combination weight rating (GCWR) of 80,000 lb.

Figure 1 shows that the baseline truck is equipped with a single vertical exhaust system. The exhaust piping consists of

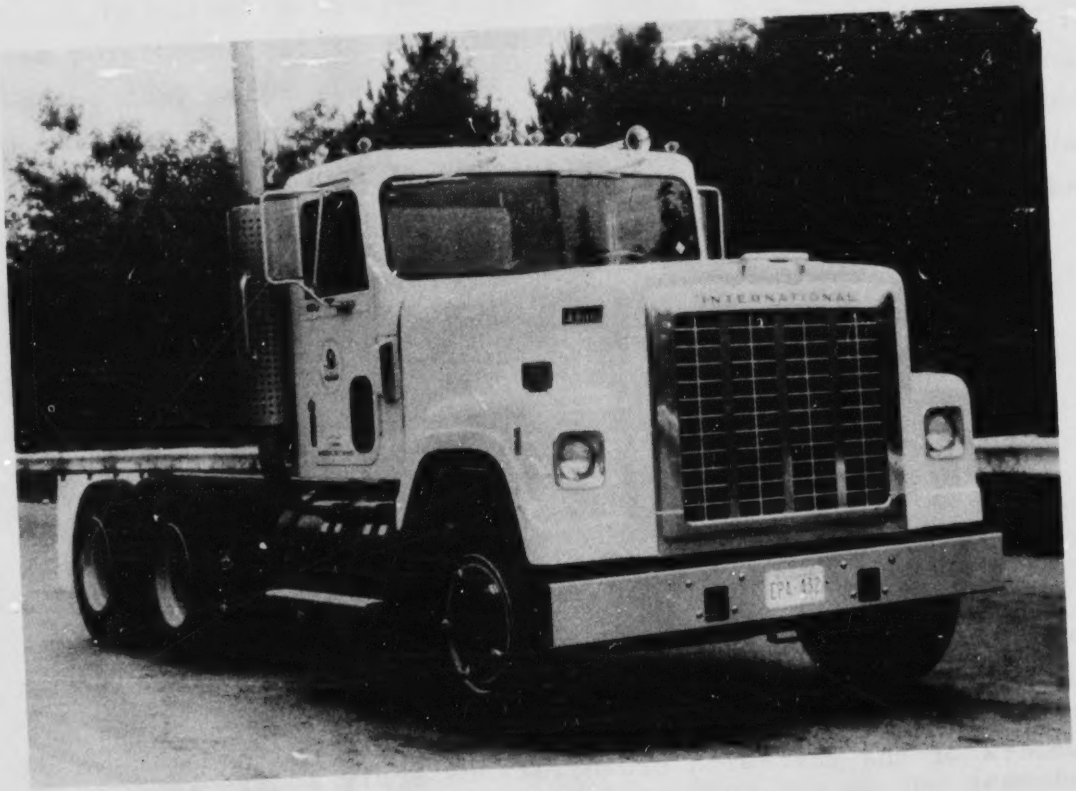

FIG. 1. BASELINE TRUCK CONFIGURATION.

sections of 5-in.-diameter stainless steel flex hose and aluminized steel tubing. The exhaust muffler, Donaldson Model MPM09-0345, has a nominal 9-in.-diameter unwrapped body and a standard 44-1/2-in. body length.

The engine, part of which is visible in Fig. 2, is a Cummins Model NTC-350 BC diesel. It is a four-stroke-cycle I-6 direct injection engine equipped with a turbocharger. The engine has an 855-cu-in. (14-L) displacement and is rated at 350 hp at 2100 rpm.

Engine intake air enters through a duct near the lower left corner of the radiator and passes through an 11-in.-diameter

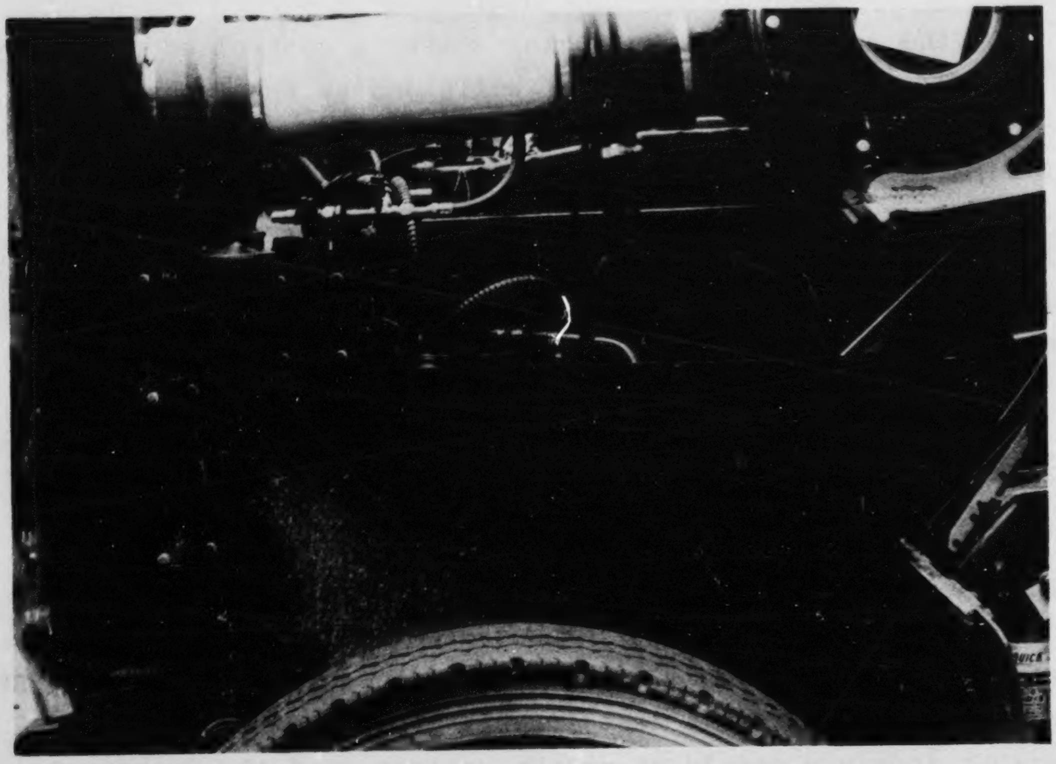

FIG. 2. LEFT SIDE OF TRUCK SHOWING MAJOR UNDERHOOD COMPONENTS.

Donaldson Model EBB16-0048 air cleaner. The air then enters the turbocharger, where it is compressed before entering the engine cylinders.

The 28-in.-diameter cooling fan has eight evenly spaced stamped sheet metal blades and is thermostatically controlled. The radiator has a frontal area of 1478 sq in. The transmission is a Fuller (division of the Eaton Corp.) Model RTF-1110 and has 10 forward speeds. The tandem drive rear axles have a 3.73 speed ratio.

All wheels were equipped with Goodyear Unisteel II 11 × 27.5 radial tires with ribbed tread patterns. These tires were selected for their noise levels, which are lower than those of the crossbar tread commonly used on tractor drive axles.

On the baseline truck, engine noise is controlled primarily by shields that fit in the wheel wells and by sound-absorptive material applied to the firewall. Figure 2 shows the left shield, which serves the dual purpose of splash protection and noise reduction. A closer view of the right shield in Fig. 3 illustrates that it is bolted to the radiator support bracket at the front and connected by means of a spring to the cab. Figure 3 also shows the attachment of sections of 1-in. fiberglass to the firewall.

2.2 Baseline Noise Levels

The truck was initially noise-tested by EPA at its Noise Enforcement Facility at Sandusky, Ohio, and subsequently by BBN at Hanscom Field in Bedford, Massachusetts. Both tests were performed in accordance with the test procedure prescribed by EPA in 40 CFR 205 [6]. This test is very much like the SAE J366b test; it involves accelerating the vehicle at full throttle from

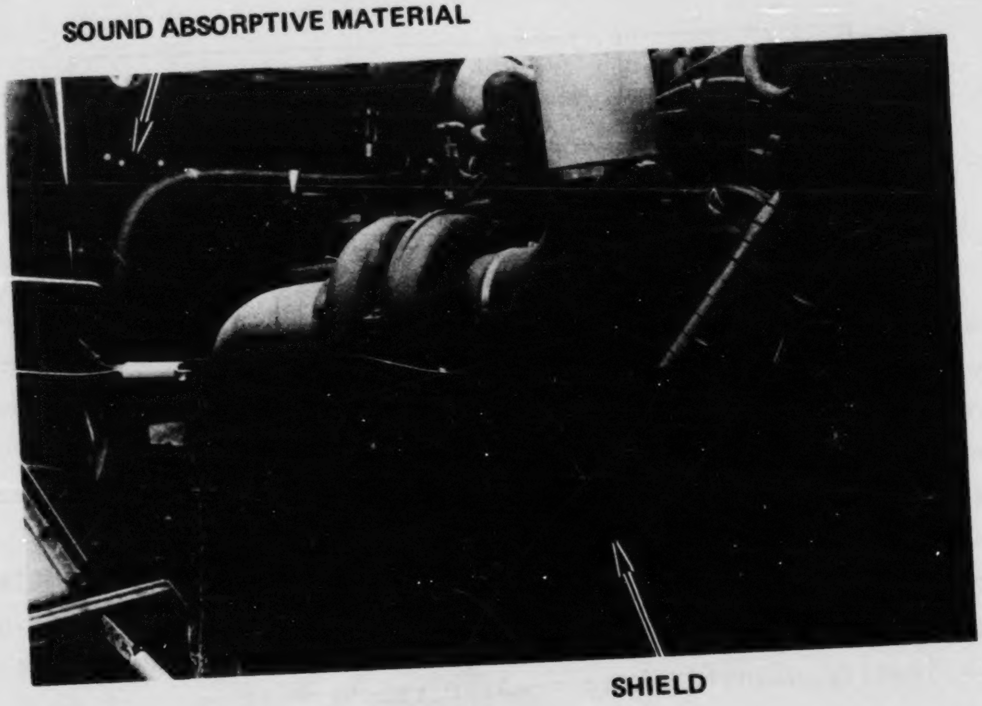

FIG. 3. RIGHT SIDE OF TRUCK SHOWING MAJOR NOISE CONTROL COMPONENTS.

an initial low speed (of about 11 mph for this truck) to a final speed at which maximum governed speed is reached. Noise levels are measured by a microphone located 50 ft from the vehicle's line of travel.

Table 1 shows that the exterior noise levels measured at each location are within about 1 to 2 dBA of each other. We will use 81.1 dBA as the baseline level for consistency with most of the tests conducted by BBN.

It is useful to know the approximate initial contributions of major noise sources on which to base the design of noise

TABLE 1. BASELINE OVERALL NOISE LEVELS.

	EPA Measurements (dBA)	BBN Measurements (dBA)
Left Side	79.2	81.1
Right Side	79.4	79.5

treatments. Laboratory and field tests were conducted to determine the contributions from exhaust, intake, engine and transmission, and tire and aerodynamic sources. However, it should be remembered that while these levels provide guidelines for the development of noise treatments, they are of only secondary importance to the levels of the treated components and complete truck. Therefore, we seek reasonable levels of accuracy (e.g., ±2 dBA) and do not feel that greater precision for these tests would justify significantly greater resource investment than is reported here.

Intake Noise

The baseline intake noise level was measured under laboratory conditions at the Donaldson Company's facility. The experimental configuration is shown in Fig. 4. The laboratory consists of an area inside a building, housing a test engine and dynamometer, and an outdoor area in which key components and a microphone are located. The acoustic wall shown in the figure is part of the building and is constructed of a double wall of concrete and an exterior foam surface. The concrete is sufficiently thick to attenuate noise radiated by the engine to negligibly low levels. The sound-absorbing foam is intended to minimize the contribution of intake noise that is reflected from the concrete wall. The EBB16-0048 air cleaner and air intake duct used in

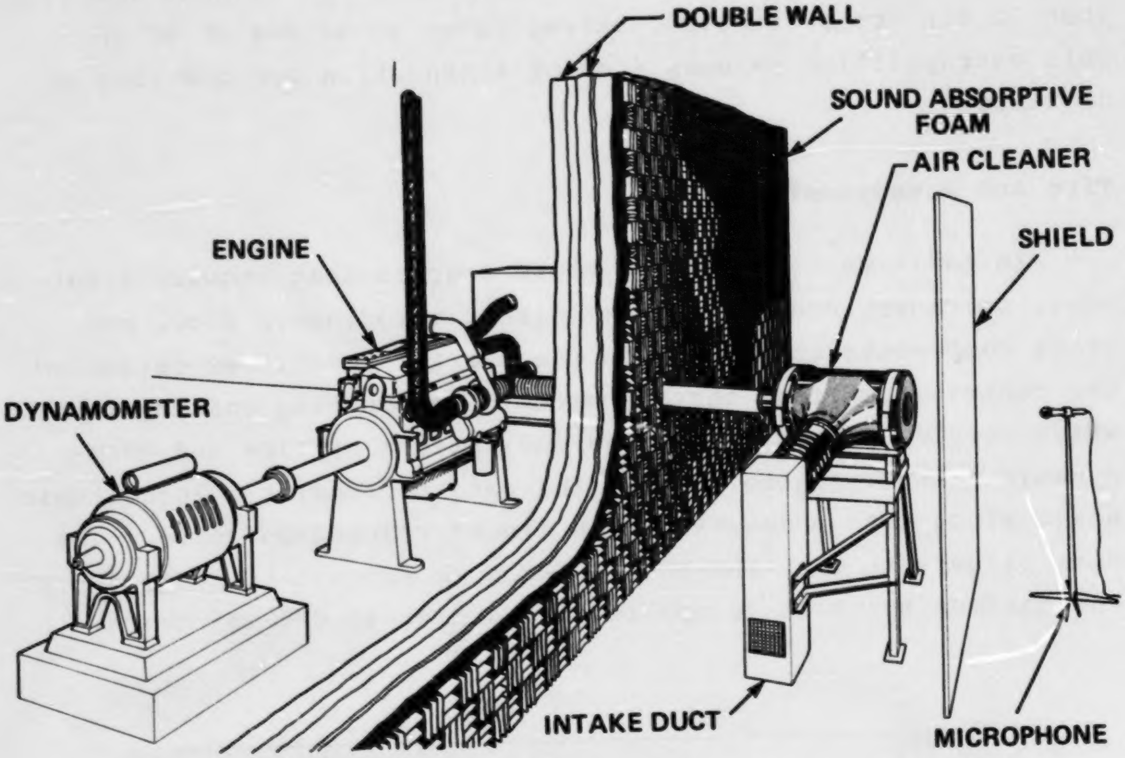
DOUBLE WALL
SOUND ABSORPTIVE FOAM
AIR CLEANER
SHIELD
ENGINE
DYNAMOMETER
INTAKE DUCT
MICROPHONE

FIG. 4. EXPERIMENTAL CONFIGURATION FOR INTAKE NOISE MEASUREMENT.

the test are the same models as those installed in the F-4370. A metal shield was placed between the intake and the microphone, as shown in Fig. 4, to simulate the effect of the hood on the radiated sound field.

Because intake noise levels were relatively low, a microphone was placed 75 in. from the intake duct so that an adequate signal-to-noise ratio could be obtained. To simulate the operational conditions that occur during a truck passby test, the engine is accelerated, using only the rotary inertia of the dynamometer as a load. (Donaldson has found that levels measured by this techniuqe correlate well with passby measurements.) The noise level measured under these conditions was 65 dBA, which, when 18 dBA are subtracted, extrapolates to 47 dBA at 50 ft. This extrapolation assumes 6 dB of attenuation per doubling of distance.

Tire and Aerodynamic Noise

In addition to the major noise sources that require treatment, secondary sources such as tires, aerodynamic flow, and other components contribute to the overall level. We estimated the contribution from these sources by conducting coastby tests, which provide particularly good indications of tire and aerodynamic noise. Figure 5 shows the data plotted on a logarithmic scale along with a least-squares linear regression curve. The data illustrate that the contribution is approximately 60 dBA at the maximum speed of 20 mph reached during 40 CFR 205 tests.

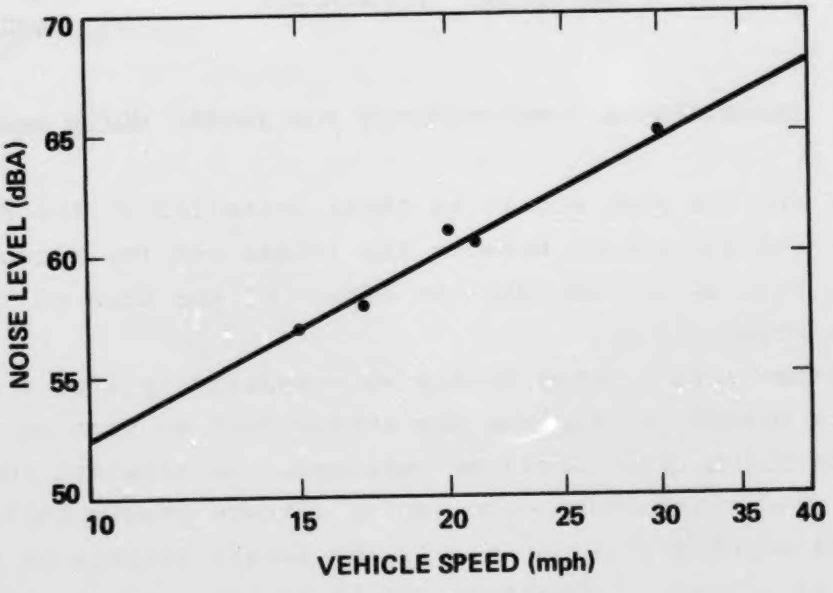

FIG. 5. VEHICLE COASTBY LEVELS.

Exhaust Noise

Estimates of the exhaust noise levels were developed from laboratory tests conducted as described above for intake noise measurements. For exhaust noise tests, however, the microphone was located 50 ft from the exhaust stack. The peak level was 74 dBA, which occurred during a runup test. As indicated earlier, the results of this type of test correlate well, but not exactly, with vehicle passby test levels.

Engine and Transmission Noise

For this project, the engine and transmission are treated as a single source, around which an acoustical enclosure is to be built. The noise contribution from the engine/transmission combination is estimated by logarithmically subtracting the levels of the other major known sources (exhaust, intake, tires and aerodynamic) from the measured overall level of 81.1 dBA. The resulting 80.1-dBA level shows that the engine/transmission level is very close to the overall level and is the dominant source of noise.

2.3 Summary of Component Levels

Figure 6 provides an overview of the major noise source levels for the vehicle in its initial, or baseline, configuration and the goals for the treated sources. The figure clearly shows the dominance of the engine and transmission, with the exhaust second and the intake, tires, and aerodynamic sources at significantly lower levels. The goals reflect some judgment as to the feasibility, reasonableness, and costs of silencing each source.

The state of the art of flow silencers is sufficiently well developed to make 60 dBA a reasonable goal for the exhaust systems. Achieving 14 dBA of additional exhaust noise reduction,

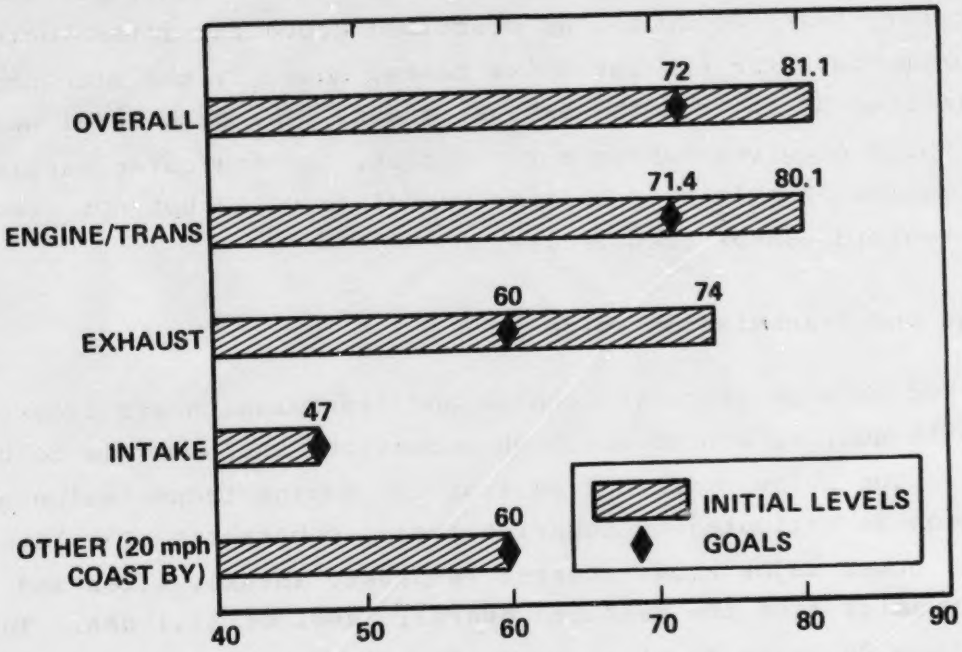

FIG. 6. OVERVIEW OF MAJOR NOISE SOURCE LEVELS AND GOALS.

though substantial, is believed feasible with a dual system incorporating off-the-shelf equipment. The initial intake noise level of 47 dBA requires no further treatment. Reducing coastby noise beyond the present 60-dBA level would have little effect on the total truck noise level associated with the low-speed test used in this program. Moreover, it would probably require tire development, which could be extensive and is beyond the scope of this effort.

3. NOISE CONTROL TREATMENTS

Three major treatments were used to reduce the noise of the International Harvester F-4370 truck. The treatments are:

- Modifications to the exhaust system
- Installation of an engine/transmission enclosure
- Installation of two-stage engine mounts.

Sections 3.1, 3.2, and 3.3 describe these treatments in detail.

3.1 Exhaust System

The dual exhaust system installed on the vehicle is shown in Figs. 7 and 8. Its components are the same as those used in the Ford CLT 9000 and GM Brigadier [3,4]. A 5-in.-diameter exhaust

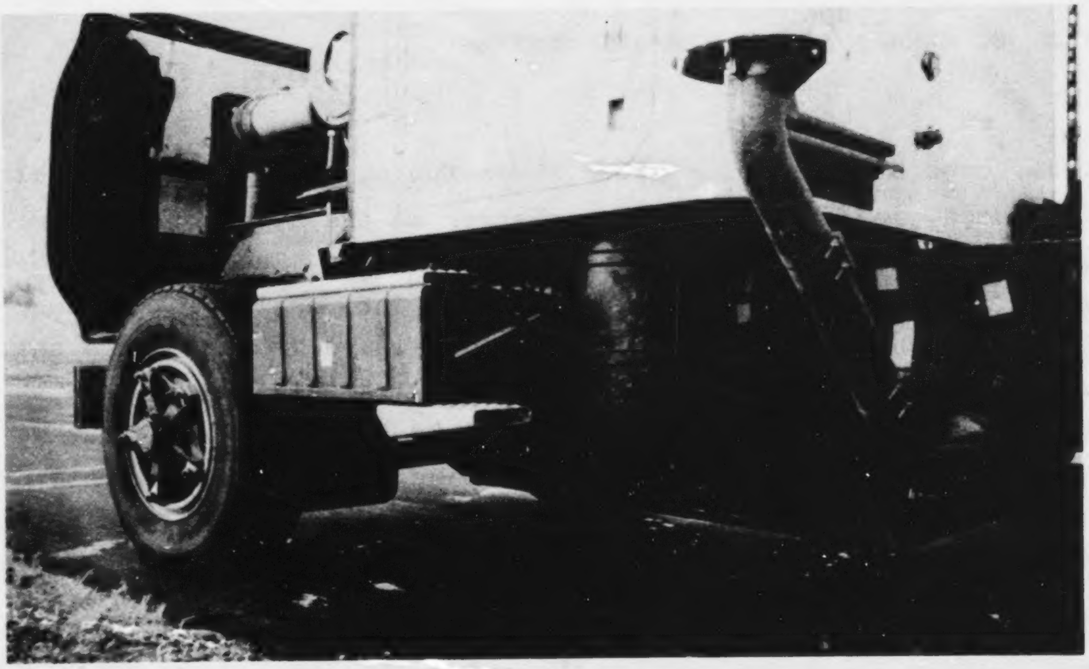

FIG. 7. CLOSEUP VIEW OF EXHAUST SYSTEM.

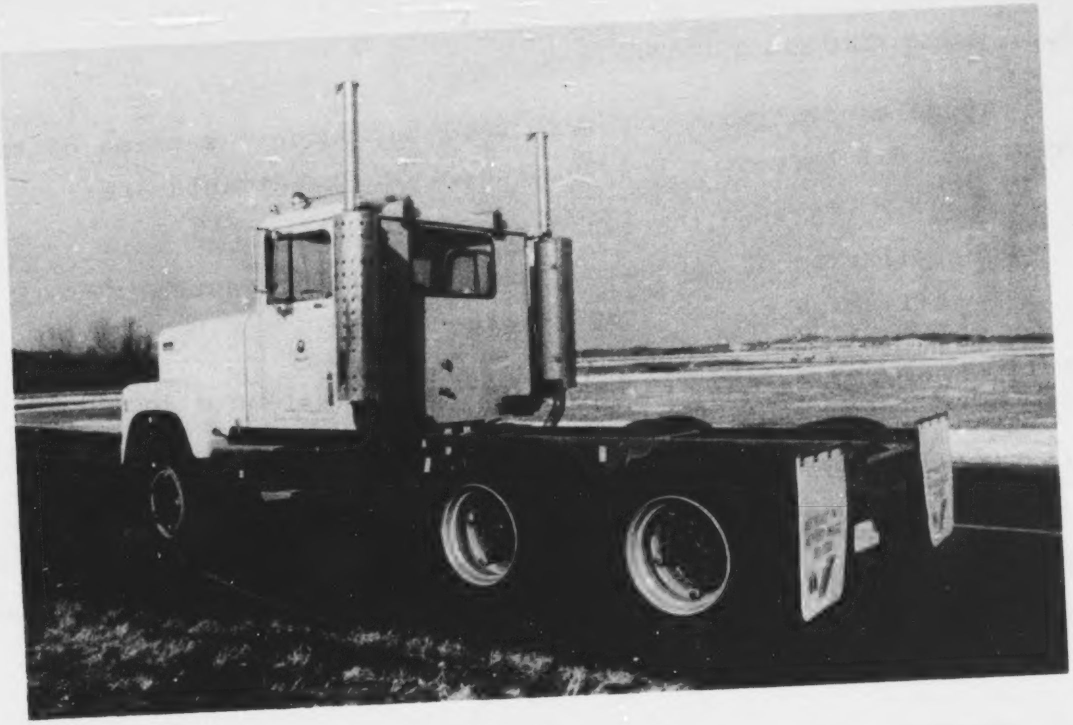

FIG. 8. REAR VIEW OF EXHAUST SYSTEM.

line, consisting of aluminized steel tubing and stainless steel flex hose, leads from the turbocharger to the Splitter Tee Can (Donaldson Model MAM10-0059) shown in Fig. 7. The Tee Can provides some muffling and splits the flow into dual 4-in. exhaust lines. Each line contains a nominal 10-in.-diameter double shell cylindrical muffler (Donaldson Model WTM10-0066)* and a 4-in. stack silencer (Donaldson Model AEM00-1337). The Super Stack Silencer, as it is designated by Donaldson, has a 3-in.-diameter perforated liner made of aluminized steel, fiberglass packing, and a pressure recovery cone at the outlet. Note that it was necessary to add a stock IH exhaust stack bracket to the left side of the vehicle to accommodate the dual system. As illustrated in Fig. 8, each 10-in. muffler is covered by a perforated heat shield.

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*The mufflers used on the truck were the bright stainless steel versions of this model.

Noise Levels

The exhaust noise level is substantially below the overall truck level and cannot be measured readily during a passby test. Accordingly, an indirect measurement must be made and the results used to estimate the passby contribution. We have used two such measurements. One is based on laboratory tests and the other on truck measurements with a microphone located close to the exhaust line terminus.

The laboratory tests were conducted with a single branch of the exhaust system located outside of the dynamometer test facility used for intake noise measurements described in Sec. 2. The other branch and the intake were remotely located and heavily silenced so that they would not contribute significantly to measured levels. The engine was run up at full throttle to governed rpm and the A-weighted level recorded. The results, illustrated in Fig. 9, show that the peak level of 58.6 dBA is reached at approximately 2060 rpm. Subtracting the 52-dBA ambient level and adding 2 dBA to account for the presence of dual exhaust gives an estimated truck exhaust noise level of 59.5 dBA. (A 2-dBA correction, rather than the 3 dBA that one might expect from elementary theoretical considerations, has been found empirically to account well for the additional branch in a dual system.)

An analysis of the spectrum of the runup sound level was also performed. In this case the runup was performed twice for each standard octave band from 63 to 8000 Hz, and the peak level

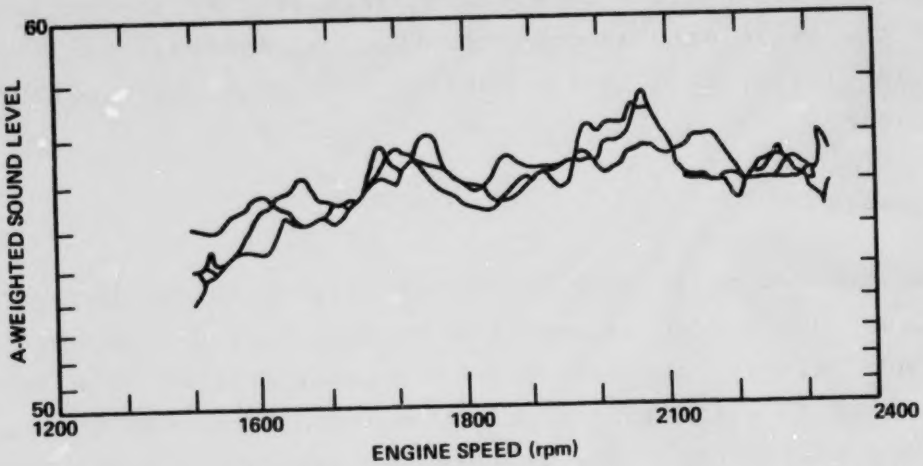


FIG. 9. EXHAUST SYSTEM NOISE LEVELS DURING RUNUP TESTS.

was read from a sound level meter with an integral octave band filter. Adding 2 dBA to the average levels to account for the presence of two exhaust lines on the truck gives the A-weighted octave band spectrum shown in Fig. 10.

Noise levels were also measured by means of a microphone located 18 in. outboard of the centerline of an exhaust system branch. Extrapolating the level measured at this location to the 50-ft microphone is best done empirically, because of ground reflections and the fact that the propagation path changes constantly during the test. An empirical relation between the level measured at 18 in. and the level measured at 50 ft was found in a separate test. A straight stack was installed on the vehicle to obtain an exhaust-dominated level at both microphones. The difference between the one-third octave band spectra for both signals gives the transfer function relating the sound at the far microphone to the sound at the near microphone. This transfer function is given in Fig. 11.

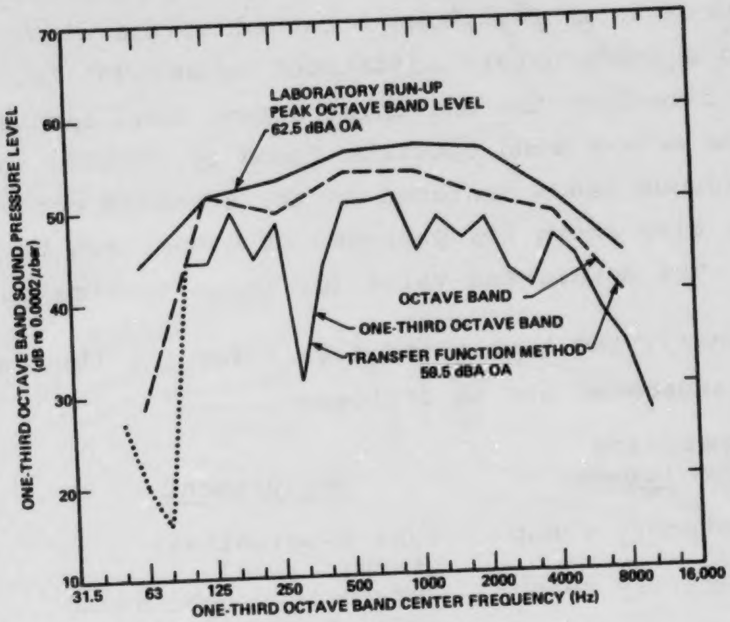


FIG. 10. ESTIMATED EXHAUST NOISE SPECTRA AT 50 FT. (Data includes a 2-dBA correction for the presence of dual exhaust lines.)

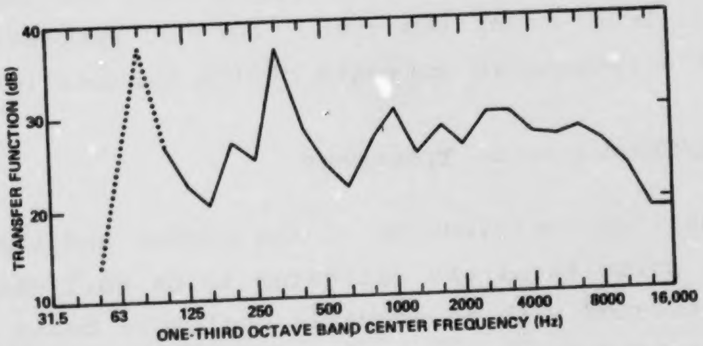


FIG. 11. TRANSFER FUNCTION FOR IH F-4370 EXHAUST.

From the transfer function in Fig. 11, a spectrum of the sound measured at a microphone attached to the final exhaust system, and a 2-dBA upward adjustment to account for the two stacks, we calculate the one-third octave band spectrum shown in Fig. 10. An octave band spectrum found by summing the levels in three contiguous bands centered on the standard octave band frequencies is also shown for purposes of comparison to the laboratory data. The A-weighted value for these spectra is 59.6 dB.

In summary, the A-weighted levels for the three different types of measurement are as follows:

Operating Conditions	Measurement	Level (dBA)
1. Laboratory runup	Peak A-weighting: graphic level	59.5
2. Laboratory runup	Peak octave band sound level meter (fast)	62.5
3. Truck passby	18 in. extapolated to 50 ft	59.6

The passby and peak A-weighted levels agree very well. The sum of the peak octave band levels is higher than the other levels. This is not unexpected because the peaks would occur at different times in the run-up cycle, are not additive, and the fast setting on the sound level meter was used. For our purposes we regard 59.5 dBA as a reasonable estimate of the exhaust level.

3.2 Engine/Transmission Treatment

The baseline contribution of the engine and transmission to the overall noise level was estimated to be 80.1 dBA. This source was treated with an acoustic enclosure built around the engine/transmission to control airborne noise. In addition, special two-stage engine mounts were installed to control structureborne sound radiation. Both treatments are illustrated in Fig. 12.

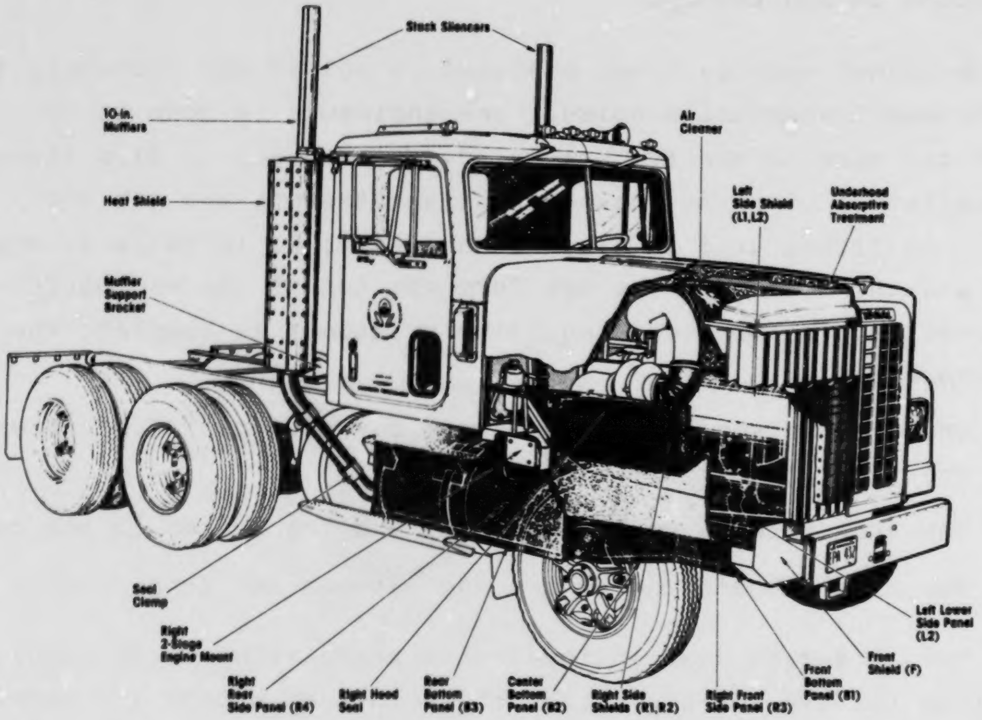

Stack Silencers
10-in. Mufflers
Heat Shield
Muffler Support Bracket
Air Cleaner
Left Side Shield (L1,L2)
Underhood Absorptive Treatment
Seal Clamp
Right 2-Stage Engine Mount
Right Rear Side Panel (R4)
Right Hood Seal
Rear Bottom Panel (B3)
Center Bottom Panel (B2)
Right Side Shields (R1,R2)
Right Front Side Panel (R3)
Front Bottom Panel (B1)
Front Shield (F)
Left Lower Side Panel (L2)

FIG. 12. NOISE CONTROL TREATMENTS INSTALLED ON IH F-4370.

The following overall design objectives guided the design of the enclosure:

- Adequate noise reduction
- Minimal effect on engine cooling performance
- Minimal maintenance interference
- Simplicity and ease of construction
- Durability
- Protection of sound-absorptive material from environmental contaminants
- Light weight.

Enclosure Design Concept

A tunnel enclosure was designed to shield the community from engine and transmission noise. The enclosure is open at the front and rear of the truck to allow cooling air to flow through the radiator, over the engine and transmission, and out the rear. As illustrated in Fig. 12 and described in Table 2, the hood and the bottom of the cab form the top of the enclosure. The remaining major areas requiring treatment to complete the enclosure are:

- The area between each frame rail and the inner fenders of the fiberglass hood
- The area between each frame rail and the bottom of the cab
- The area beneath the engine and between the frame rails.

The IH F-4370 came equipped with heavy rubber side shields to block the line of sight from the roadside through the wheel wells to the engine. That type of treatment was not adequate for the level of engine-noise reduction required here. Consequently, the side shields were removed and replaced with panels L1, R1, L2, and R2. These panels are attached to the frame rail and together seal the space between the inner fenders and the frame rail from the radiator to the firewall.

Below the frame rails, panels L3 and R3 form the side walls of the bellypan forward of the firewall. Aft of the firewall to the back of the cab, panels L4 and R4 perform the same function. Panels B1, B2, B3, and F close the bottom of the bellypan from the radiator to the back of the cab.

The gap between the bottom of the cab and the frame rails aft of the firewall is sealed with a 1/8-in. thick sheet of rubber. These gap shields extend from the back of the cab forward to the firewall on both sides of the cab.

TABLE 2. DESCRIPTION OF ENCLOSURE NOISE TREATMENTS.

Designation	Description
L1, R1	Left and right forward side shields above the frame rail
L2, R2	Left and right aft side shields between the firewall and L1 and R1
L3, R3	Left and right side panels of the bellypan forward of the firewall
L4, R4	Left and right side panels of the bellypan between the firewall and the back of the cab
B1, B2, B3	Panels forming the bottom of the bellypan
F	One-piece enclosure sealing the space between the bottom of the radiator and panel B1

Except as noted above, the enclosure is fabricated primarily from sheet aluminum. While it is anticipated that a truck manufacturer would use an alternate material (e.g., sheet steel), sheet aluminum provides a light, rigid material well suited to prototype work. A minimum panel thickness of 1/8 in. was dictated by requirements for strength and durability rather than for noise reduction. This 1/8-in. aluminum panel thickness is more than adequate to provide the required noise reduction [2].

Sound-Absorptive Material

Three types of absorptive treatments were used in the enclosure:

- IH baseline 1-in. fiberglass
- BBN-installed 1.5-in. Mylar-wrapped fiberglass behind perforated aluminum sheet metal
- BBN-installed 2-in. unprotected fiberglass.

The IH-installed absorptive treatment is found only on the firewall. This material was left undisturbed. The 1.5-in. Mylar-wrapped fiberglass was attached to panels L4 and R4 on each side of the transmission below the frame rails, and to panels L1, L2, R1, and R2 on each side of the engine above the frame rails. Figure 13 shows the absorptive treatment on panel R1, and Fig. 14 shows the treatment on panel L1. As Fig. 14 shows, 100% coverage of these panels was not possible, because we had to allow for penetration of these panels by such components as the steering wheel shaft, as shown in the figure. This type of absorptive treatment and its acoustic performance have already been described elsewhere [3].

The 2-in.-thick unprotected fiberglass is installed on the inner surface of the hood above the frame rails (Fig. 15), and on the underside of the cab floor above the transmission. These are areas that, because of their remoteness, are unlikely to receive much mechanical damage. In addition, they tend to be high up in the enclosure where contamination by water and oil is less of a problem. Accordingly, it was decided to forego the use of perforated metal for mechanical protection and the use of Mylar wrapping to prevent contamination in these areas.

Side Shelves (R1, L1, R2, and L2)

The two side shields and the hood form the enclosure forward of the firewall and above the frame rails. Each side shield is formed of two separately removable panels. Figure 14 shows the left side shield composed of panels L1 and L2.

The Cummins NTC-350 engine in the F-4370 was 2 to 3 dBA noisier than the engines in two other trucks previously quieted in the Demonstration Truck Program [3,4]. Accordingly, a higher

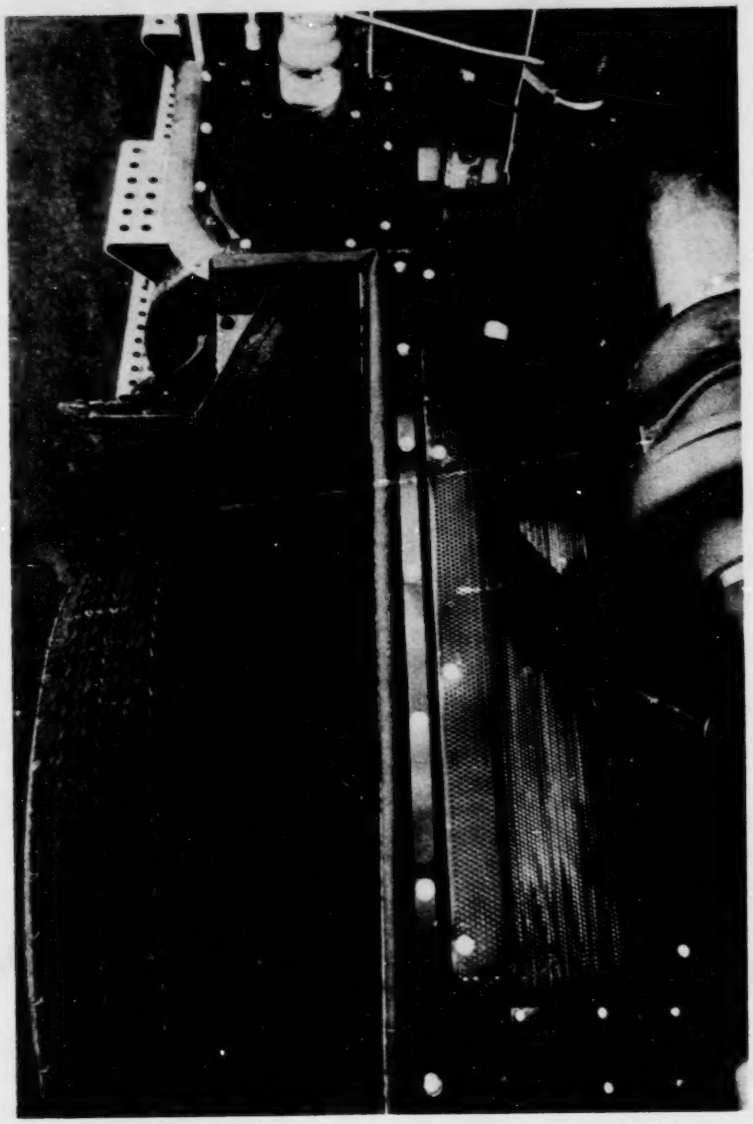


FIG. 13. ABSORPTIVE TREATMENT ON PANEL R1 AS SEEN FROM FRONT, LOOKING AFT.

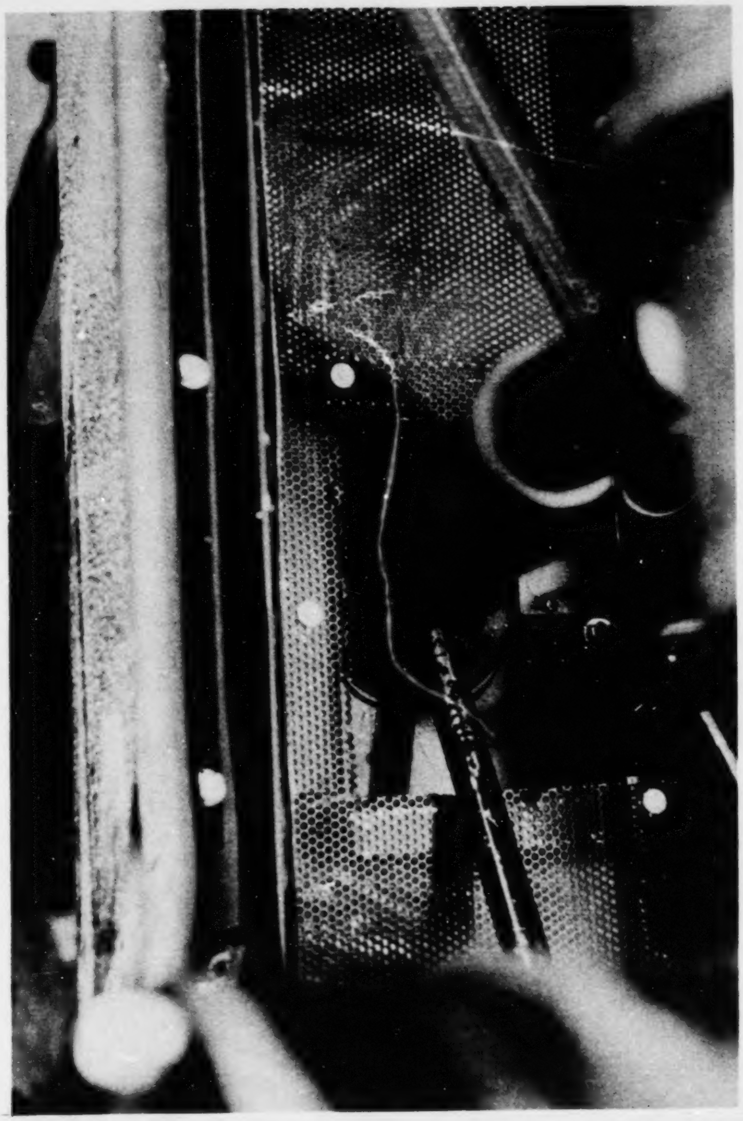


FIG. 14. ABSORPTIVE TREATMENT ON PANEL L1 AS SEEN FROM ABOVE. (Note cutout for steering wheel shaft.)

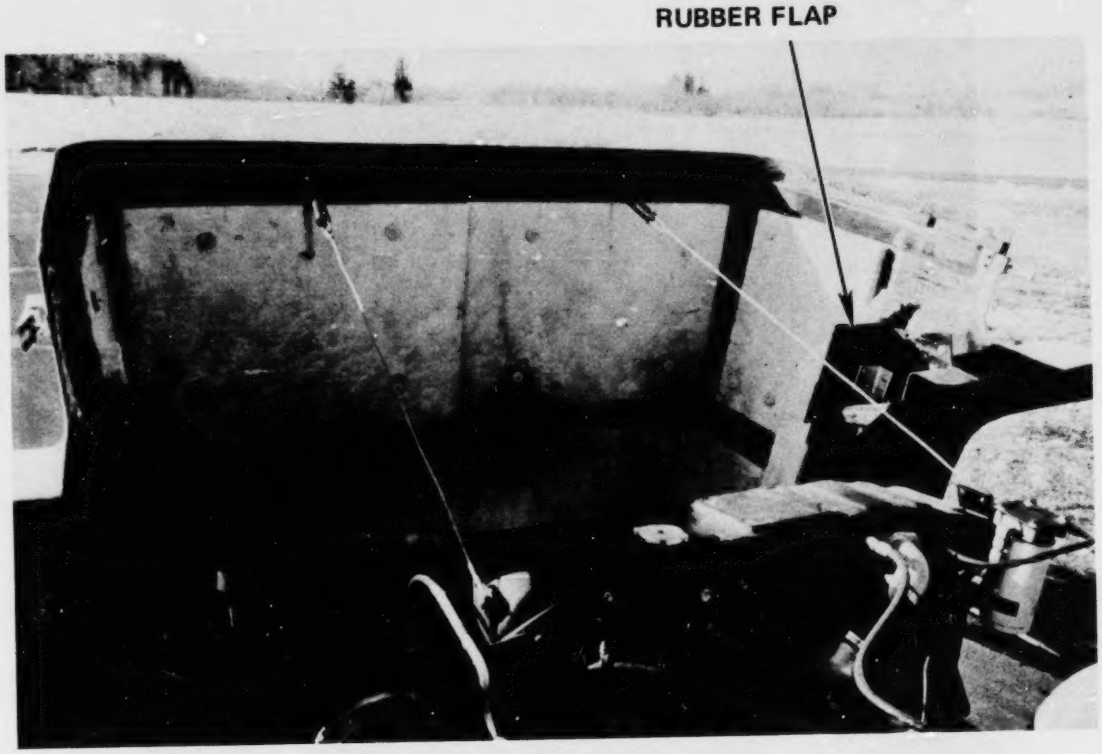


FIG. 15. UNPROTECTED ABSORPTIVE TREATMENT IN HOOD AREA.

insertion loss was required from this enclosure than from the earlier enclosures in the Ford CLT 9000 and the GMC Brigadier. To achieve the higher insertion loss, we took great care to seal all openings in the enclosure to the maximum practical extent possible. Of course, the opening through the radiator and the opening at the rear of the enclosure were retained to allow for the passage of cooling air. One element in the sealing of the enclosure is the rubber "P-seal" shown in Figs. 16 and 17, and so named because of its shape in cross section. The P-seal is attached to the top edge of panels L1 and R1 and seals against the inner surfaces of the two inner fenders of the fiberglass hood.

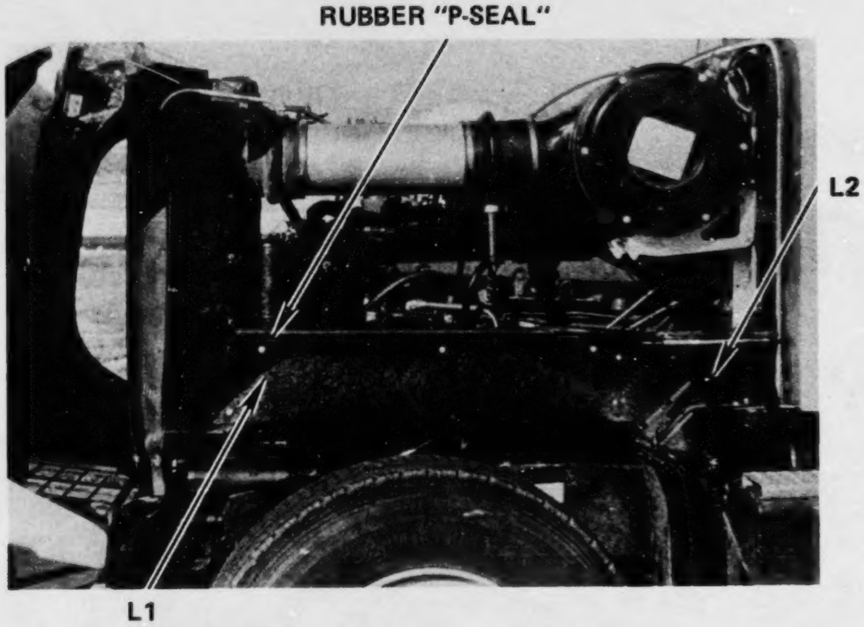

RUBBER "P-SEAL"
L2
L1

FIG. 16. LEFT SIDE SHELF.

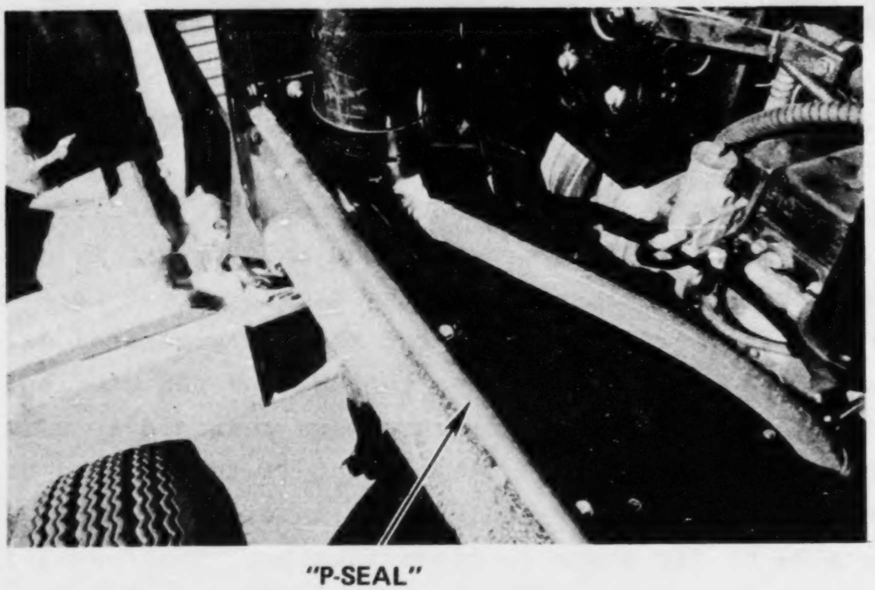


FIG. 17. PANEL L1, AS SEEN FROM THE FIREWALL, LOOKING FORWARD.

At the rear of the side shields near the firewall, the seal against the hood on both sides of the truck is accomplished by means of a wiping seal shown in Fig. 18, and by means of a rubber flap attached to the hood that seals against a shelf at the aft end of each side shield when the hood is in the closed position. The flap on the right hand side is shown in Fig. 15. The shelf against which that flap seals is shown in Fig. 18.

The truck came equipped with a rubber seal where the hood joins the cab body, as shown in Fig. 18. Despite the presence of the seal, gaps between the hood and the cab body existed when the hood was closed and latched using the IH rubber latches shown in Fig. 19. To pull the hood down into close contact with the seal at the cab body, the heavy-duty latches shown in the figure were installed. With these latches properly adjusted and closed, the hood fit tightly against the body of the cab along the full lengths of the hood seal.

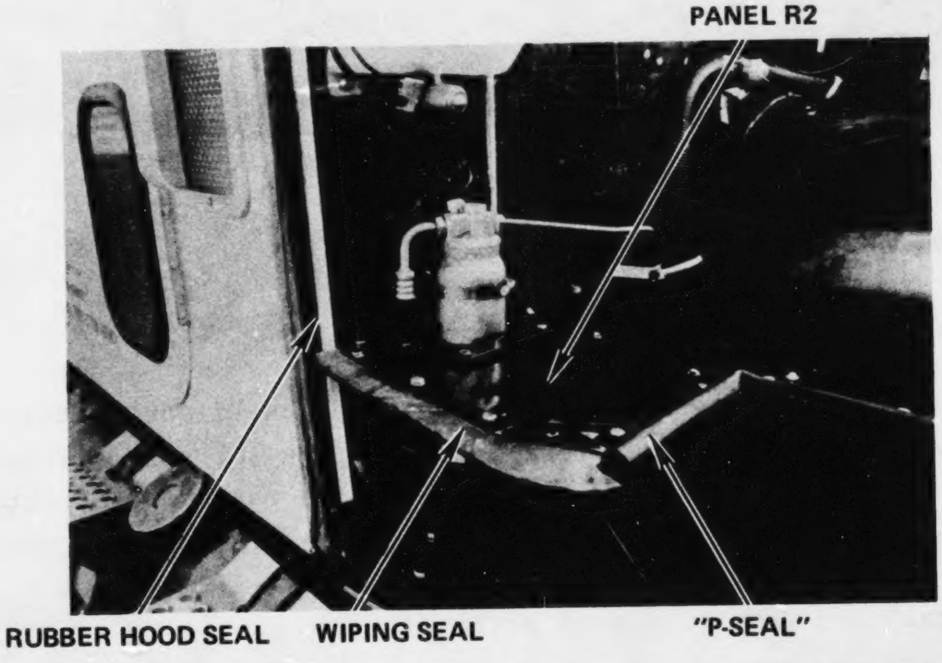


FIG. 18. RIGHT SIDE SHIELD AND FIREWALL, AS SEEN LOOKING AFT.

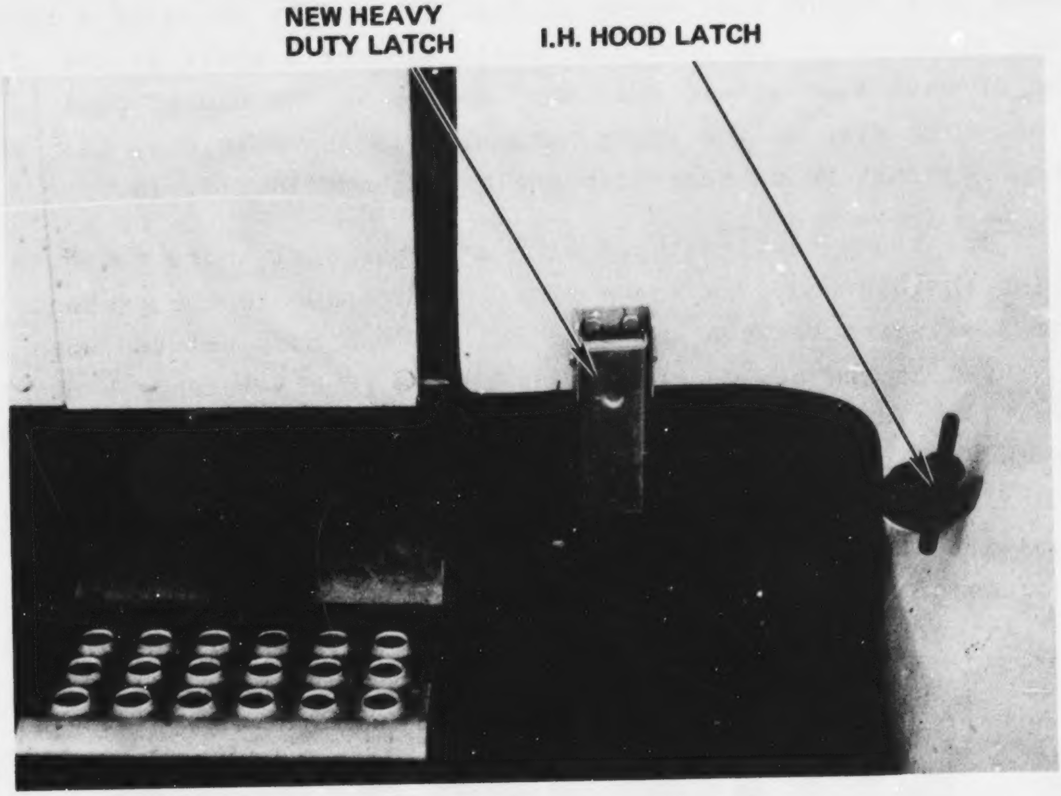


FIG. 19. HOOD LATCH SYSTEM.

Gap Shields

The gap shields fill the space between the floor of the cab and the frame rails. They were made from 1/8-in. rubber sheet with the top edge bolted to the floor of the cab and the bottom edge simply resting on the top flange of the frame rail. The rubber sheet was cut oversize so it would rest firmly on the frame rail and provide a good seal.

Bellypan (R3, L3, R4, L4, B1, B2, B3, F)

The bellypan encloses the bottom of the engine, extending from the bottom of the radiator to the rear of the cab. The design goals for the bellypan were:

- Maximum accessibility for maintenance purposes
- No reduction of ground clearance
- Quick removal and replacement of bottom panels
- Provision for drainage
- Adequate clearance over front axle.

Panels R4 and L4 are fabricated from 0.160 in. aluminum. The panels, which are attached to the frame rails with brackets, start at the bottom flange of the frame rail and extend down to form the side walls of the bellypan aft of the firewall.

Just aft of the forward spring shackles, near the firewall, panels R3 and L3 (also fabricated from 0.160-in. aluminum) attach to panels R4 and L4. R3 and L3 extend forward and attach to the trunnion, forming the side walls of the bellypan forward of the firewall. The enclosure narrows in the forward half to provide clearance for the leaf springs on each side of the enclosure as shown in Fig. 20. As a consequence of this narrowing of the enclosure, the top edges of panels R3 and L3 cannot seal against the frame rail as panels R4 and L4 do. There is, in fact, a significant gap between these panels and the frame rails that is filled with a 1/4-in.-thick rubber sheet. One edge of the rubber sheet is bolted to the side panels, and the other edge lays

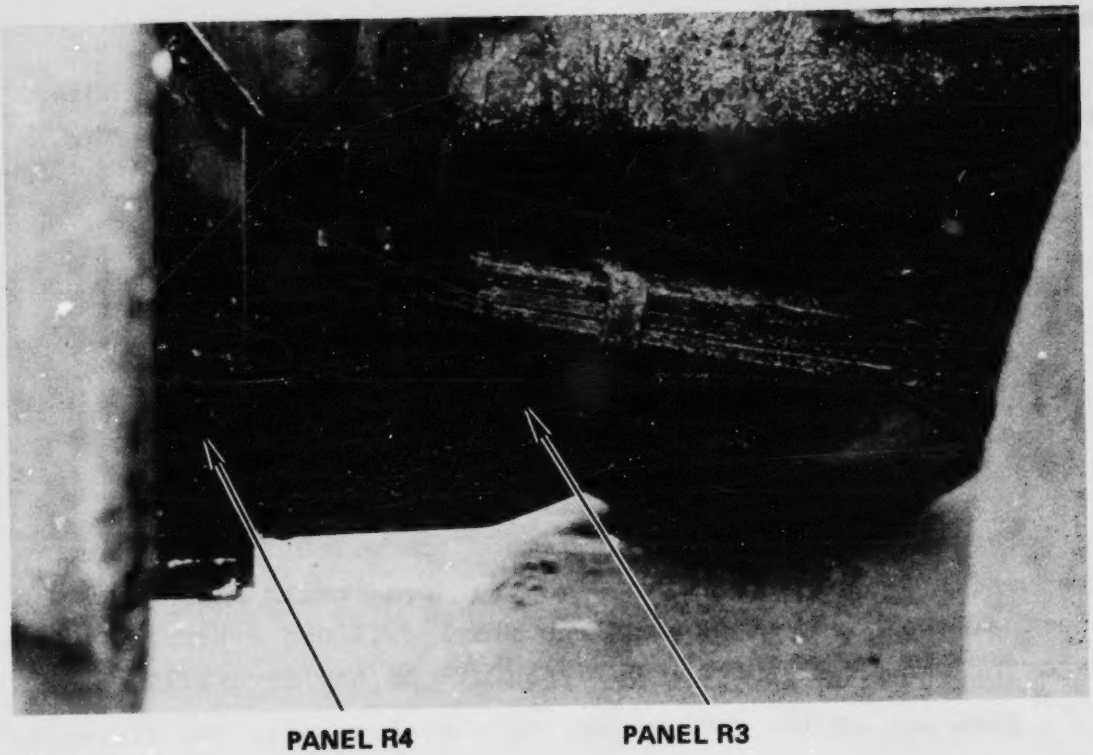


FIG. 20. JUNCTION OF BELLYPAN PANELS R3 AND R4, BELLYPAN AS SEEN FROM RIGHT SIDE.

against the top surface of the lower flange of the frame rail as shown in Fig. 21.

The bottom of the bellypan is sealed with four panels, all fabricated from 0.125-in. aluminum. Panels B1, B2, and B3 are attached to the side panels with quick release quarter turn fasteners (Southco Model No. 85). The panels are designed to be quickly and easily removed and reinstalled for routine maintenance of the engine and transmission. The remaining panel, the front shield, F, shown in Fig. 12, is a box-shaped unit that attaches to the bottom of the radiator, fits around the trunnion

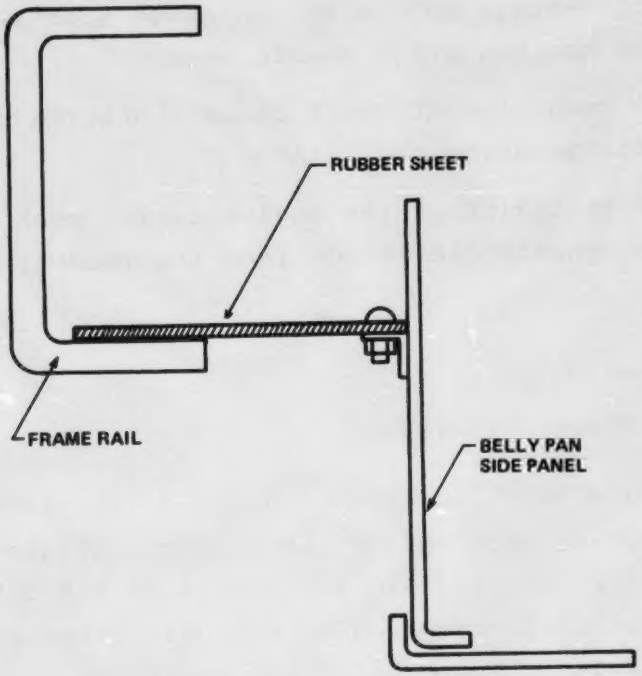


FIG. 21. SEALING ARRANGEMENT FOR FORWARD HALF OF BELLYPAN AT FRAME RAIL.

and fills the space between the bottom of the radiator and B1, the first panel at the bottom of the bellypan.

3.3 Two-Stage Engine Mounts

It was discovered early in the program that structureborne vibration from the engine and transmission, while not a dominant noise source in the untreated IH F-4370, could be a significant contributor after exhaust noise and engine/transmission airborne noise were reduced. Past experience has shown that significant reduction in engine/transmission structureborne noise from heavy-duty diesel trucks can usually be obtained by improving only the two rear engine mounts [1]. The approach chosen to decrease the transmission of vibration through these mounts was to convert them from single-stage mounts to two-stage mounts. The design objectives for the two stage mounts were:

- Adequate reduction of truck frame vibration caused by engine/transmission excitation
- Adequate restraint of the engine during peak torque operation and dynamic excitation from the roadway
- Durability
- Simplicity
- Minimum weight penalty.

As illustrated schematically in Fig 22, a two-stage mount incorporates a blocking mass between isolators. If the single-stage mount has been properly designed, such that its deflection under dynamic load is large compared to the deflection of the frame rail at the mounting point, then the insertion loss due to the use of a two-stage mount can be readily calculated. The calculation shows that the increase in vibration isolation is given by

$$IL = 20 \log \frac{2K1}{K_2} \frac{1}{1 - \left(\frac{\omega}{\omega_0}\right)^2} \tag{1}$$

where K_2 and K_1 are the stiffness of the two-stage and single-stage mount isolators, respectively, and ω_0 is the resonant frequency of the blocking mass on the isolators. This expression applies only if the engine and frame rail mounting points are rigid. The insertion loss, calculated using this expression, is illustrated schematically in Fig. 22 under the assumption that the same isolators were used in both single- and two-stage mounts. Around the resonant frequency ω_0, the two-stage mount actually transmits more vibration than a single-stage mount.

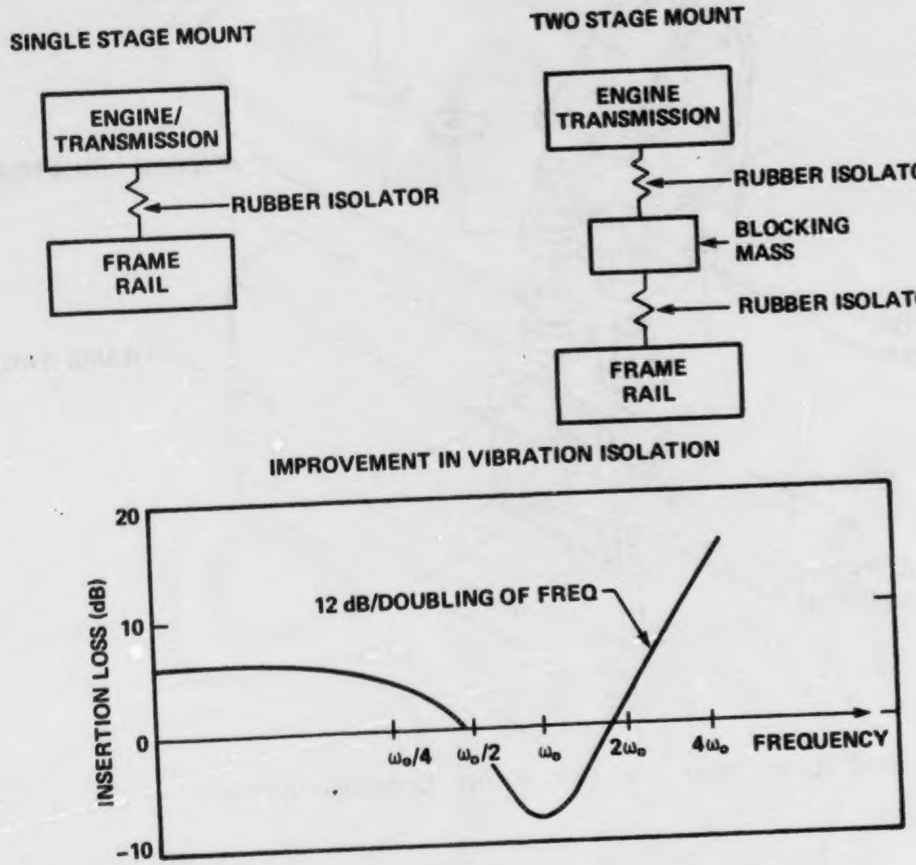

SINGLE STAGE MOUNT
ENGINE/ TRANSMISSION
RUBBER ISOLATOR
FRAME RAIL
TWO STAGE MOUNT
ENGINE TRANSMISSION
RUBBER ISOLATOR
BLOCKING MASS
RUBBER ISOLATOR
FRAME RAIL
IMPROVEMENT IN VIBRATION ISOLATION
INSERTION LOSS (dB)
20
10
0
-10
12 dB/DOUBLING OF FREQ
ω₀/4
ω₀/2
ω₀
2ω₀
4ω₀
FREQUENCY

FIG. 22. SCHEMATIC ILLUSTRATION OF SINGLE AND TWO-STAGE ENGINE MOUNTS.

Above ω_0 the insertion loss increases rapidly. Accordingly, one usually seeks to make ω_0 as low as possible. In practice the isolator stiffness cannot be made too small because the engine mounts must be stiff enough to support the loaded engine within its clearance envelope. Similarly, the mass cannot be made too large because of weight and space restrictions.

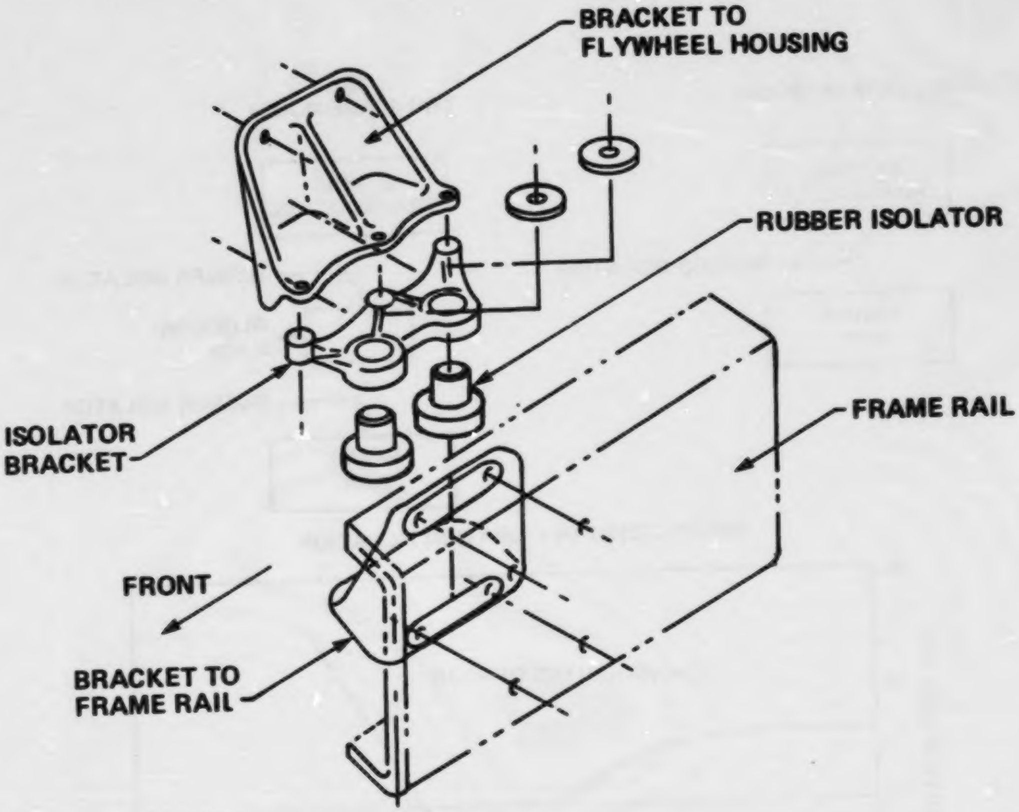

BRACKET TO FLYWHEEL HOUSING
RUBBER ISOLATOR
FRAME RAIL
ISOLATOR BRACKET
FRONT
BRACKET TO FRAME RAIL

FIG. 23. ORIGINAL REAR ENGINE MOUNT CONFIGURATION.

Fortunately, the original single-stage mounts were suitable for conversion to two-stage mounts. Figure 23 shows the geometry of the mounts before modification. The two rubber isolators are pressed into an isolator bracket. That bracket is bolted to the bottom of a second bracket, which in turn is bolted to the engine/transmission assembly at the flywheel housing. The rubber isolators rest on the top surface of a third bracket, which is bolted to the web of the frame rail. Bolts pass through the isolators, securing the engine to the frame rail.

Providing space for converting this mount to a two-stage mount required two modifications. First, the isolator bracket that bolted to the bottom surface of the flywheel housing bracket was moved and bolted to the top surface of that bracket. Second, the holes in the frame rail bracket were enlarged to accept the rubber isolators. A 12-lb steel block, the largest that could be accommodated, was then fabricated to fit in the resulting space and act as the blocking mass. The same types of isolators as those used in the original single-stage mount were used here, two above the mass in the isolator bracket and two below it in the frame rail bracket. Bolts passed through the isolators into tapped holes in the mass. The assembly is shown in the photograph of Fig. 24.

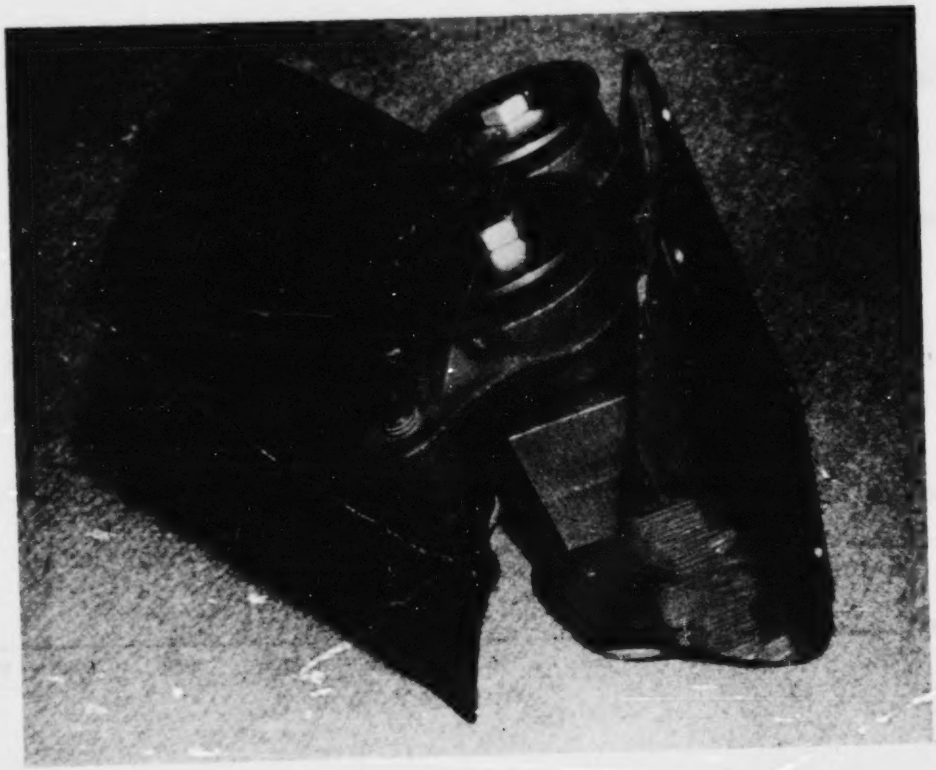


FIG. 24. TWO-STAGE ENGINE MOUNT.

The ratio of the vertical vibration on the engine side of the mount to the vertical vibration on the frame rail side of the mount was measured before and after installation of the two-stage engine mounts. Figure 25 shows the increase in that ratio because of the two-stage mount. As the figure clearly shows, except in the one-third octave bands at 400 and 500 Hz, the two-stage mount significantly decreases the transmission of vibration from the engine to the frame rail.

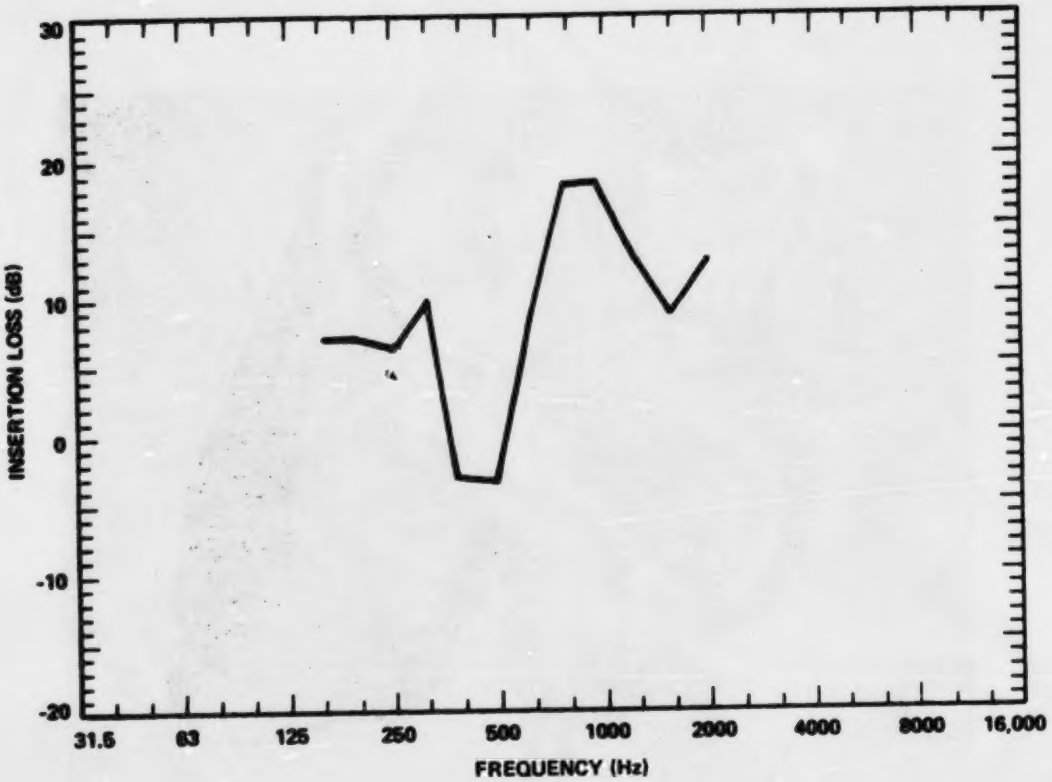


FIG. 25. INCREASED VIBRATION ISOLATION CAUSED BY TWO-STAGE ENGINE MOUNT.

Static Test of the Two-Stage Mount

To ensure the safe operation of the two-stage engine mounts during fleet service, we arranged with Teledyne Engineering Services, Waltham, Massachusetts, to carry out a static load test on one mount. Figure 26 shows the mount placed between two specially fabricated fixtures in Teledyne's MTS electrohydraulic test machine. During the test, the load was gradually increased while load and deflection were simultaneously recorded on an X-Y plotter. Figure 27 shows the trace of the force and deflection as recorded during the test. The small dips in the curve at 4000, 10,000, 15,000 and 18,000 lb are a result of stopping the increase in deflection to photograph and examine the mount. During that time the rubber in the mount relaxed the load causing the dip.

The load was increased to a maximum of 19,600 lb, where the mount failed. One of the three bolts holding the isolator bracket to the flywheel housing bracket (see Fig. 23) broke, causing the failure. In fact, all three bolts, as well as the isolator bracket, began to bend at 15,000 lb. However, only the one bolt failed. Figure 28 shows a closeup of the mount at zero load and at 18,000 lb. The severe deformation of the mount at the high load is readily apparent.

International Harvester designs its engine mounts assuming a 3g cyclic load plus the load from the stall torque of the engine. For the Cummins NTC 250, the static load is 1098 lb per mount (3294 lb for a 3g cyclic load) and the stall torque loads each mount by 4380 lb. The IH design load is then 7674 lb. This is well below the 19,600 lb at which the mount failed and almost a factor of two below the 15,000 lb load at which bending deformation in the mount became evident. On the basis of the IH criterion, the mount is more than adequately designed for fleet service.

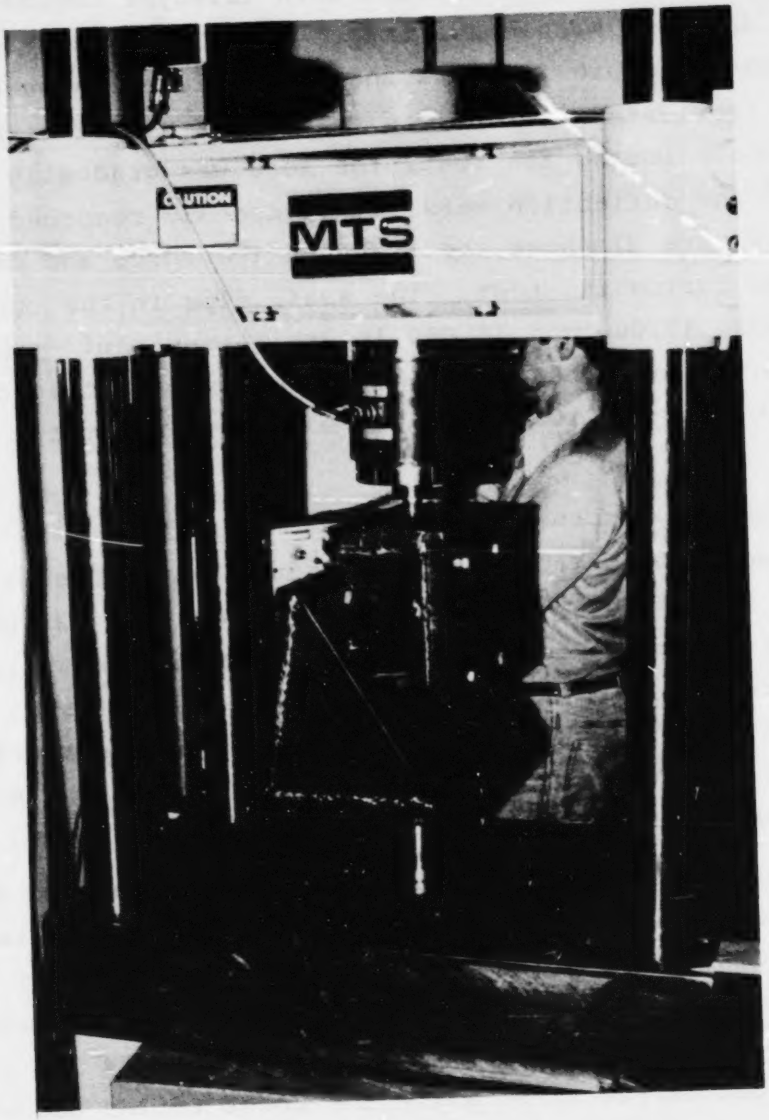

CAUTION
MTS

FIG. 26. STATIC LOAD TEST SETUP FOR TWO-STAGE ENGINE MOUNT.

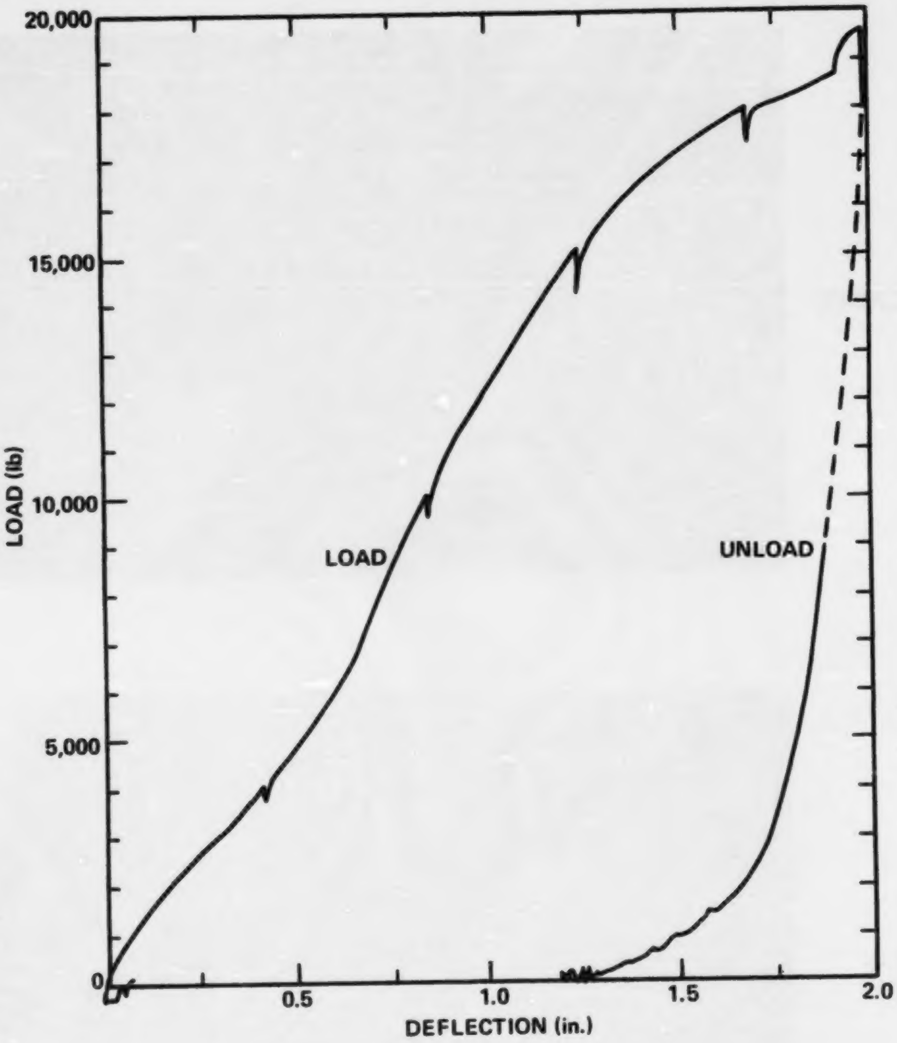
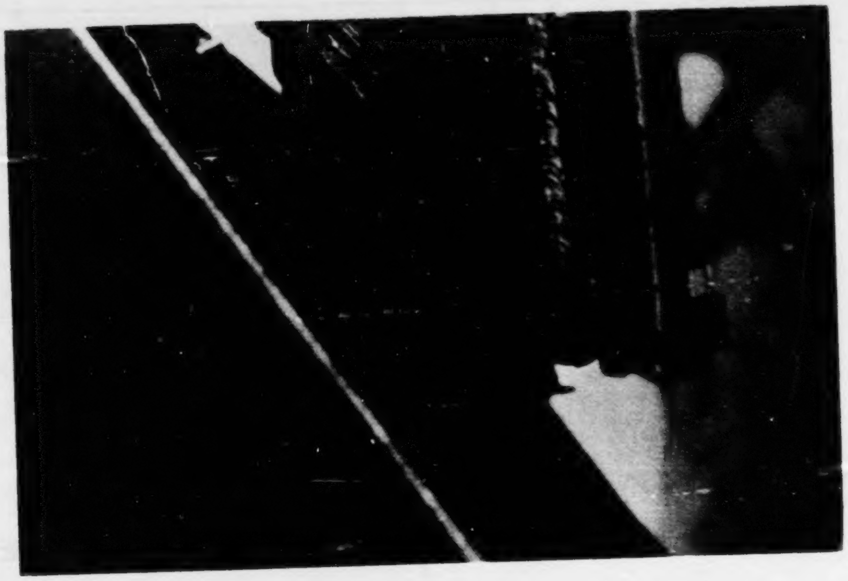
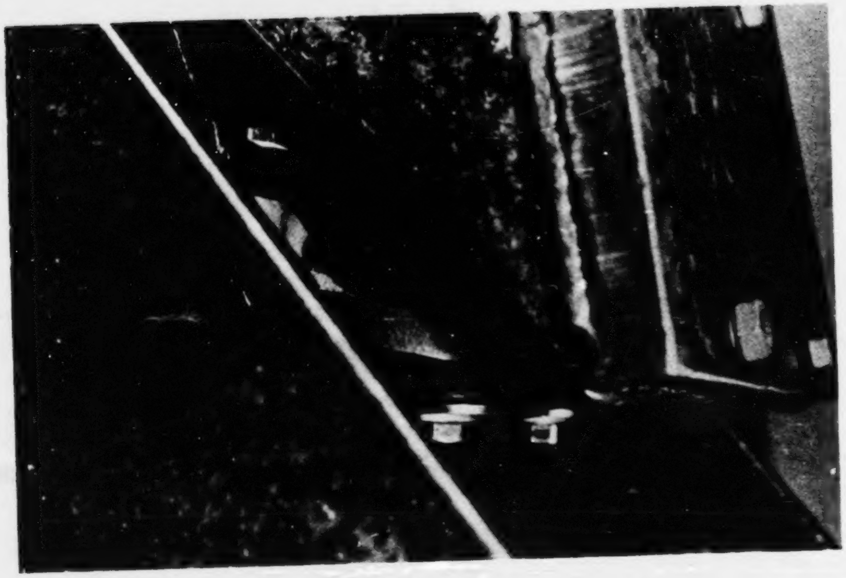


FIG. 27. FORCE DEFLECTION CURVE FOR THE TWO-STAGE ENGINE MOUNT.



a) UNLOADED



b) 18000 lb LOAD

FIG. 28. DEFORMATION OF THE TWO-STAGE ENGINE MOUNT UNDER LOAD.

4. FINAL NOISE LEVELS

Measurements of exterior and interior noise levels were conducted according to the procedures described in Appendix A of this report. The results are reported here.

4.1 Exterior Noise Levels

Table 3 summarizes the noise source contributions for the initial and final configurations. An 8.4-dBA reduction in overall vehicle noise was achieved. The tunnel enclosure reduced the airborne contribution from the engine and transmission to the overall noise level by 8.9 dBA, to 71 dBA. The structureborne contribution from the engine and transmission, with the enclosure, was estimated to be 67.8 dBA. The two-stage engine mounts were used to support the engine only at the two rear mounting points. The single-stage rubber mount at the front of the engine was not changed. The two-stage mounts reduced the engine/transmission structureborne noise by 2.6 dBA, to 65.2 dBA. The two treatments together reduce overall engine noise by 7.8 dBA, resulting in a treated engine/transmission source contribution of 72.3 dBA.

TABLE 3. SUMMARY OF NOISE SOURCE CONTRIBUTIONS.

Source	Initial Level dBA	Final Level dBA	Noise Reduction dBA
Engine/Transmission	80.1	72.3	7.8
• Airborne	79.9	71.0	8.9
• Structureborne	67.8	65.2	2.6
Exhaust	74.0	59.5	14.5
Intake	47.0	47.0	-
Other (coastby)	60.0	60.0	-
Total	81.1	72.7	8.4

Exterior noise levels were measured by BBN in Cambridge, Massachusetts, on January 27, 1981 and by General Motors in association with a conference held at their facility. The results, shown in Table 4, are in reasonable agreement with each other.

TABLE 4. FINAL EXTERIOR NOISE LEVELS.

	BBN Measurements Cambridge, MA			GMC Measurements		
	Run 1	Run 2	40 CFR 205 Level	Run 1	Run 2	40 CFR 205 Level
Left Side	72.5	72.8	72.7	71.3	71.6	71.5
Right Side	72.2	72.1		71.5	71.2	

4.2 Interior Noise Levels

Figure 29 shows the SAE J336a criteria [7] and the octave-band interior noise levels measured after the application of noise treatment. The criteria band levels shown in Fig. 29 are those that are summed to establish an overall criterion against which actual levels are to be compared. The maximum allowable band levels, established by the SAE J336a Recommended Practice, are not to be exceeded if the vehicle is to meet the design criteria.

The truck meets the design criteria in that the sum of the measured band levels, 98.6 dB (84.9 dBA), is less than the sum of the criteria band levels, 102.9 dB (87.6 dBA).

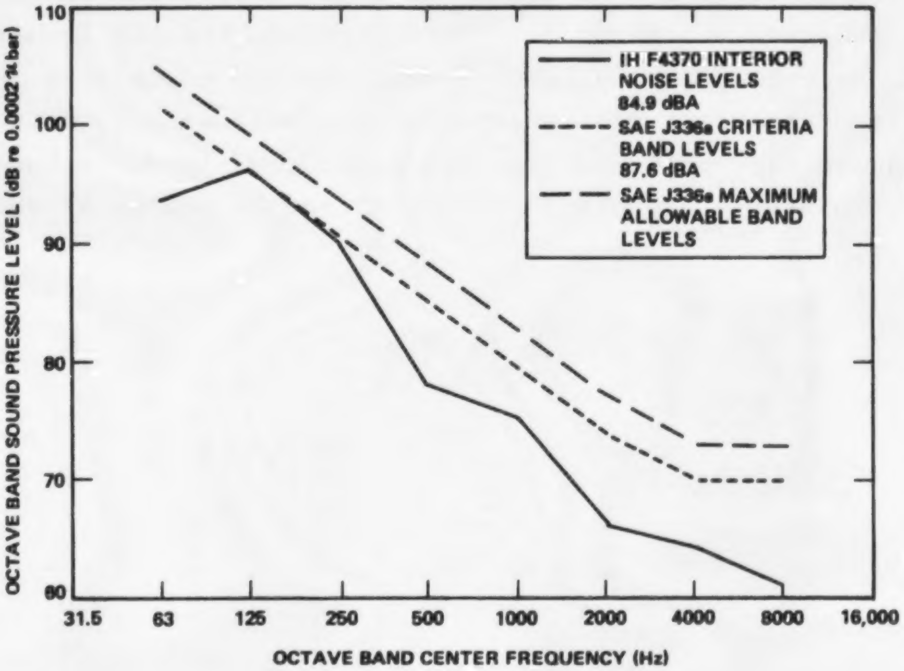


FIG. 29. TRUCK INTERIOR NOISE LEVELS MEASURED ACCORDING TO THE SAE J336a TEST PROCEDURE.

5. COOLING PERFORMANCE

Cooling tests were conducted in the Cummins Engine Co. test facility, illustrated in Figs. 30 through 32. Air introduced by a blower in front of the truck, as shown partly masked in Fig. 31, flows over the vehicle. During a test, the air is maintained at a constant speed and temperature, and the truck runs on a chassis dynamometer with heavy chains positioning both sets of the tandem rear wheels on the dynamometer rollers. Exhaust gases from both stacks are piped outside of the facility, as shown in Fig. 32.

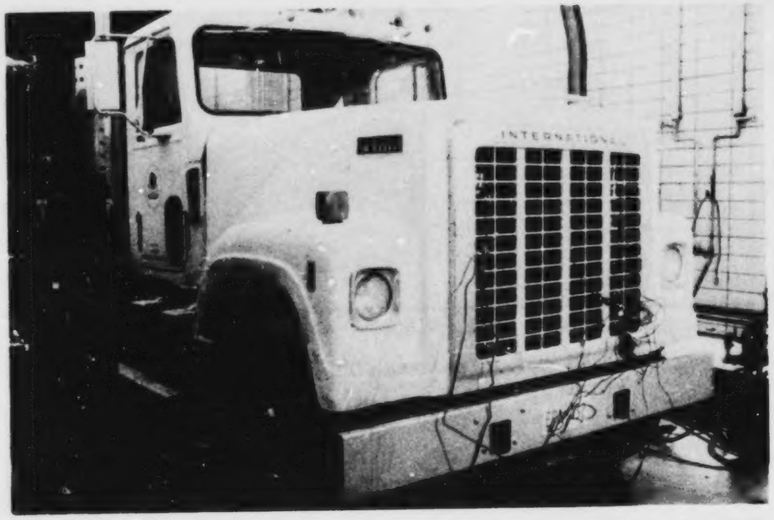


FIG. 30. FRONT VIEW OF IH F-4370 IN CUMMINS TEST FACILITY.

The primary purpose of the test is to evaluate engine cooling system performance, which is measured by the Air-to-Boil (ATB) temperature, the estimated ambient air temperature at which the coolant would reach 212°F. That is,

$$ATB = 212 - T_i + T_a \quad , \tag{2}$$

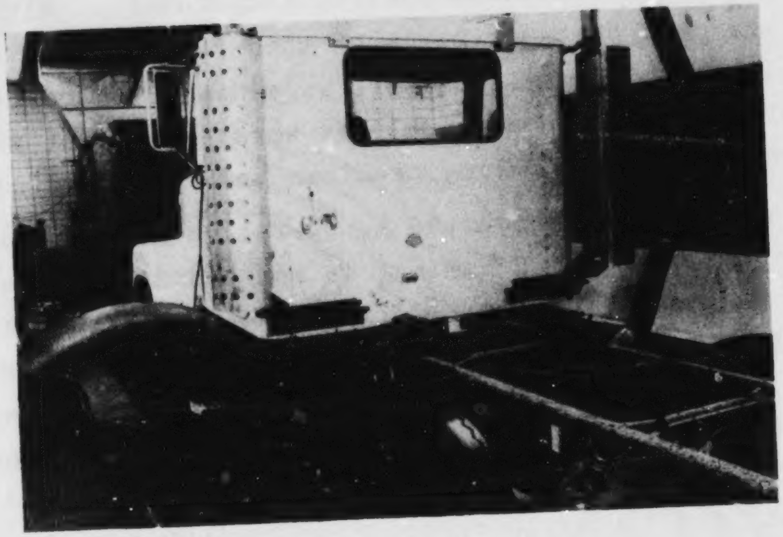


FIG. 31. VIEW TOWARD THE BLOWER IN CUMMINS TEST FACILITY.

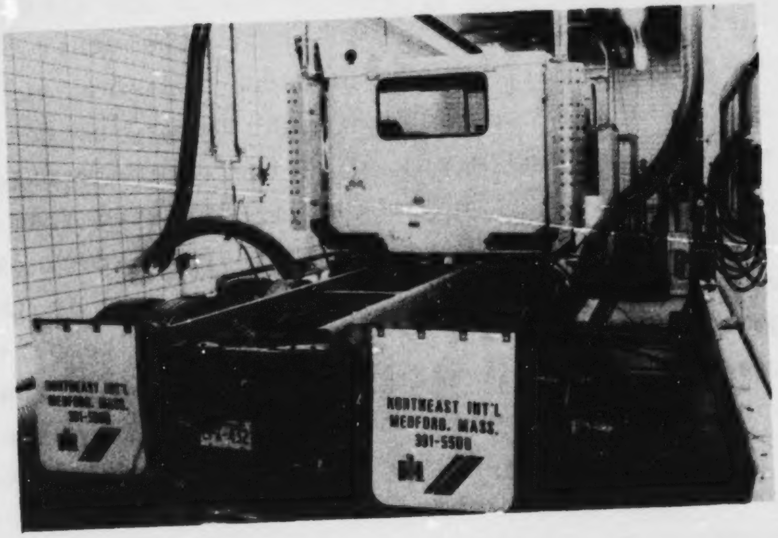

NORTHEAST INT'L
MEDFORD, MASS.
391-5500

FIG. 32. REAR VIEW OF IH F-4370 IN CUMMINS TEST FACILITY.

where T_i is the coolant temperature measured at the radiator inlet and T_a is the measured ambient temperature. Although pure water at standard pressure boils at this temperature, truck coolants operating under pressure boil at a higher temperature. Accordingly, vehicles that meet this worst-case test are very unlikely to encounter cooling problems under service conditions.

The ATB test was conducted by operating the vehicle in an ambient wind flowing at a nominal 15 mph and 80°F. The hub on the thermostatically controlled fan clutch was locked to ensure that the fan was operating, and the cab air conditioner was turned on to produce the heat that would normally be rejected by the condenser in front of the radiator. Tests were conducted on March 5, 1981 with the engine running at governed speed (2100 rpm) and at peak torque (corresponding to 1500 rpm) conditions.

The truck was first tested in its fully quieted condition. As testing progressed, it soon became clear that recirculation was occurring around the edges of the radiator, i.e., very high inlet air temperatures, 151°F, were noted in the upper left hand quadrant (driver's side) of the radiator. The other quadrants were 60°F to 70°F cooler. This indicated that hot air from the engine compartment was escaping through the gap between the radiator and the hood and mixing with the cool air entering the radiator. To alleviate this problem, we inserted foam rubber in that gap at the upper left quadrant of the radiator. The result was a reduction of radiator air inlet temperature in that quadrant to 137°F and an increase of 4°F in the ATB temperature at governed speed. As a final test, we removed the bottom panels from the enclosure and removed the foam rubber from the radiator/hood gap. Time was not available at the facility to remove entirely the engine/transmission enclosure to obtain a true baseline ATB temperature. However, the change in temperatures after removal of the bottom panels does give some indication of the effect of the enclosure on engine cooling.

The results of these tests are shown in Table 5. The ATB temperature of 108°F at rated speed is somewhat low, but after partially fixing the recirculation problem the ATB went up 4°F to 112°F. This is nearly the same as the Ford CLT 9000 [3] and the same as the specification for the GM Brigadier [4]. Examining the radiator air inlet temperatures in the four quadrants of the radiator after sealing the gap between radiator and hood at the upper left quadrant, we find, for the truck operating at governed speed and maximum power,

Upper left	137°F
Upper right	108°F
Lower left	82°F
Lower right	94°F.

Clearly, there is still considerable recirculation that could be improved by additional sealing with resulting improvement in the ATB temperature.

Removal of the bottom panels of the enclosure resulted in an increase in the ATB temperature to 115°F. Operation of the truck at peak torque generally decreased ATB temperature by 8 to 9°F. Although there is no specification of engine oil temperatures for this test, Cummins specifies 180° to 225°F as the normal operating range and 250°F as acceptable for short periods of time.

TABLE 5. COOLING PERFORMANCE OF IH F-4370.

		Rated Engine Speed		Peak Torque	
	Fully Quieted	Fully Quieted Radiator With Left Quadrant Gap Sealed	Bottom Panels Removed	Bottom Fully Quieted	Panels Removed
Air Speed (mph)	15	15	15	15	15
Ambient Air Temperature (°F)	80	80	80	80	80
Engine Speed (rpm)	2090	2090	2090	1500	1500
Gear	8	8	8	9	9
Vehicle Speed (mph)	39	39	39	36	36
Dyno Power (hp)	265	264	262	247	254
Engine Coolant Out (°F)	184	180	177	194	188
Air-to-Boil (°F)					
Measured	108	112	115	98	104
Specified	122	122	122	112	112
Engine Oil (°F)					
Measured	228	224	222	234	229
Specified*	250	250	250	250	250

*Specified by Cummins as acceptable for short periods of time.

6. COST ESTIMATES

This section contains a discussion of the costs of the noise control treatments described in previous sections. There is a specific cost attributable to the manufacture and installation of each major noise control treatment: the engine/transmission enclosure, the two-stage engine mounts, and the modified exhaust system. We first present a summary of these costs, and then discuss the procedures used to estimate the cost of each treatment. The cost of operating the vehicle, as affected by changes in fuel consumption, available payload, and maintenance, is also important and will be treated in operating reports during the in-service test program.

Table 6 presents the distinctions between costs and price used in this report. The convention is that the seller sells at a price, and a buyer buys at a cost. There are three sellers: the manufacturer of noise control products (e.g., a muffler manufacturer), the truck manufacturer, and the truck dealer. The three buyers are the the truck manufacturer, the truck dealer, and the truck operator. A markup is applied in moving from one level to another. Hence,

manufacturer's price × dealer markup = dealer's price.

TABLE 6. SUMMARY OF COSTS AND PRICES.

Transaction	Cost	Price
Sale of Component Supplier's Parts to Truck Manufacturer	Manufacturer Cost	Supplier Price
Sale of Truck by Manufacturer to Dealer	Dealer Cost	Manufacturer Price
Sale of Truck by Dealer to Operator/Customer	Operator Cost	Dealer Price

There is no single, generalized approach for cost estimation. The costing and pricing procedures of each truck manufacturer are highly confidential for competitive reasions. Our approach to cost estimation is determined largely by the treatment to be costed and the availability of information with which cost estimates can be derived. Reliance is placed on information and relationships derived in [8] and [9]. The rationale for certain assumptions is based on information presented in other reports in this series, [3,4]. All cost and price estimates are in 1979 dollars.*

6.1 Summary

Table 7 presents an overall summary of the treatment weights. Table 8 presents a summary of the estimated overall cost and price increases attributable to the noise control treatments installed on the IH F-4370. The weight of the truck increased by 332 lb, approximately 2.4% of tractor tare weight, or 0.4% of the 80,000 lb maximum permissible gross combination weight. The estimated price increase of $1,307 is a 3.2% increase over the actual purchase price of the vehicle, $40,464.

The cost and price estimates presented here are BBN estimates for the add-on treatments developed by BBN. They are not necessarily identical to the cost and price of a comparable enclosure, were it to be installed by a truck manufacturer on production level vehicles. There are reasons why BBN cost estimates could differ from actual manufacturer costs. The BBN enclosure design is essentially a tailor-made retrofit. More

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*The vehicle is a 1979 model, manufactured in June of 1979. Costs and prices are in 1979 dollars for consistency among the reports in this series.

TABLE 7. SUMMARY OF TREATMENT WEIGHTS.

Treatment	Weight (lb)	Net Increase (lb)
Engine-Transmission Enclosure		170
• Components added	180	
• Components removed	<10>	
Engine Mount Modifications		26
• Components added	26	
Exhaust System Modifications		136
• Components installed	221	
• Component removed	<85>	
Total Weight	332	332

TABLE 8. SUMMARY OF COST AND PRICE INCREASES.

Treatment	Net Increase	
	Dealer Cost ($)	Dealer Price ($)
Engine-Transmission Enclosure	460	691
Engine Mount Modifications	46	68
Exhaust System Modifications	402	543
Total	908	1302

cost-effective design and materials specification by a manufacturer for actual production vehicles might well result in different enclosure specifications and per-vehicle costs. While BBN has accounted for research, development, and testing (RD&T), and tooling costs by adjusting manufacturing cost estimates upward, that adjustment could be inaccurate, particularly if tooling or RD&T costs were atypical. The markup factors for manufacturers could differ among manufacturers from the markups assumed by BBN. Accordingly, the cost and price estimates presented here should be viewed as representative estimates for the treatments installed on the truck.

6.2 Enclosure Costs

Approach

The primary method of estimating the cost of the enclosure installed on the F-4370 was to examine the relationship between the weight of materials and the cost of materials. This is a common technique used in engineering economics. Obviously, some components, such as special machined parts and electronic devices, have a price per pound greater than the overall price per pound of the truck; others are clearly less. Our focus is the weight-cost relationship for an enclosure. The first step is to obtain data with which to estimate a relationship. Having established a relationship, we then estimate the cost of the enclosure, given the weight of the enclosure.

We have presented elsewhere [3,4] a relationship between enclosure weight and manufacturer's price, with which one can estimate the cost of an enclosure. That relationship is a least-squares regression derived from data [8]. The estimated equation is:

$$Y = 61.3 + 1.92X \qquad R^2 = 0.99 \quad , \tag{3}$$

where Y is manufacturer's price in 1979 dollars and X is enclosure weight in pounds.

The coefficient of determination, designated R^2, can be interpreted as the variation in the dependent variable (manufacturer's price) accounted for by variation in the independent variable (enclosure weight). In this instance, 99% of manufacturer's price can be "explained" by enclosure weight. The estimated slope coefficient indicates that a 1-lb increase in weight would result in approximately a $1.92 increase in manufacturer's price (or a $2.88 increase in dealer price, given an assumed markup of 1.5 in going from manufacturer's price to dealer's price.)

This equation shows only the relationship between weight and manufacturer's price of a prototype enclosure. It does not include any costs for special tooling or research, development, and testing associated with commercial production of the enclosure.* Accordingly, any cost or price estimate derived from this equation is downward biased, since it excludes these costs. Conversely, it does not reflect any cost savings attributable to production economics.

Estimated Enclosure Costs

A summary of the components and weights for each assembly of the enclosure is presented in Table 9. The assembly weights presented in the table are based on either actual weight measurements by BBN, or weight estimates derived from blueprint measurements and the weight of component material per unit area. As is evident from the table entries, the bulk of the weight increase is accounted for by fabricated aluminum components that constitute the sides and bottom of the enclosure.

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*These costs are estimated separately in the following section and added to an estimate obtained from the equation.

TABLE 9. SUMMARY OF ENCLOSURE ASSEMBLY AND COMPONENT WEIGHTS (LB).

Component	Weight
Installed	
Super Stack Silencers (2)	20.0
Mufflers (2)	122.5
Heat Shield & Bracket (1)	13.0
Tee Can (1)	19.0
4 in. Piping (100 in.)	23.0
4 in. Flex Pipe (12 in.)	3.1
Band Clamps	10.6
Mounting Bracket	10.0
Removed	
Original Muffler & Shield	<45.7>
Exhaust Stack	<14.2>
Exhaust Piping	<25.2>
Net Increase	136.1

Given the enclosure weight of 180 lb and the weight-manufacturer's price relationship presented above, the estimated manufacturer's price of the enclosure is \$407. This estimate is then incresed by 19% to account for tooling and RD&T costs. The 19% escalation applied here is the same percentage applied in earlier reports in this series [3,4]. While tooling and RD&T costs are influenced by a variety of factors, such as the complexity of the enclosure design, the materials used, and the volume of production, the 1.19 markup used in prior reports in this series has been accepted by reviewers of those reports. A 1.5 markup is then applied to manufacturer's price to obtain dealer price, estimated to be \$726. The calculations are summarized as follows:

$$\begin{array}{rll} 61.3 + 1.92(180) = & \$406.90 & \\ & \underline{\times\ 1.19} & \text{tooling and RD\&T markup} \\ & \$484.21 & \text{manufacturer's price} \\ & \underline{\times\ 1.50} & \text{dealer markup} \\ & \underline{\$726.32} & \text{dealer price} \quad . \end{array} \qquad (4)$$

The final adjustment to the estimated price of the enclosure is to credit the deletion of rubber side shield panels, which BBN removed. These panels attach to a spring at the rear of the radiator and extend aft to the firewall. Each panel is approximately 1.5 ft by 3.0 ft. The replacement part cost of the panels is \$70.60. Over-the-counter retail part prices have a high markup to cover the costs of distribution, inventory, and sales. A 100% markup is not uncommon. Hence, we estimate that the \$70.60 after-market price corresponds to a \$35.30 price of the panels on the truck as delivered. This retail price of \$35.30 corresponds to a manufacturer's price of \$23.73.

The results of this price estimation procedure for the enclosure are summarized as follows:

estimated dealer price - enclosure	$726.32	
less deletion of side shields	<35.30>	
net dealer price increase	$691.02	. (5)

6.3 Engine Mounts

BBN installed two-stage mounts on the F-4370. As described in Sec. 3, the main material difference between the standard IH mounts and the BBN mounts was a 12-lb mass that was added to each standard rear mount. Large rubber isolators and longer mounting bolts are also part of the BBN two-stage mounts.

The DOT Quiet Truck Program [8] provided cost data on two-stage mounts for the Freightliner Quiet Truck. The design of the BBN mounts for the F-4370 is essentially similar to those designed for Freightliner.* The total weight difference between the Freightliner and IH two-stage mounts, 68 lb and 26 lb respectively, partially reflects that BBN was able to incorporate the original IH mounting brackets in its two-stage mount.

The dealer price of the Freightliner two-stage mounts was $72 in 1973 dollars. This can be expressed in 1979 dollars by applying the Producer Price Index for iron and steel. This index stood at 136.2 in 1973 and rose to 283.5 in 1979, an increase of 108%. Thus, the dealer price of the Freightliner two-stage mounts, in 1979 dollars, would be $150, or $2.20/lb. A dealer markup of 1.5 is assumed in the Freightliner estimates and this implies a manufacturer's price of $1.47/lb.

Given the comparability of the Freightliner two-stage mounts and those installed by BBN on the F-4370, we applied these dollar-per-lb estimates, $1.47 and $2.20, to the incremental 26-lb of the BBN mounts. We also applied a 1.19 markup to cover tooling and RD&T. BBN estimates the incremental manufacturer's price of the two-stage mounts to be $45, and the incremental dealer's price to be $68. This estimate implies a price-per-lb, including tooling and RD&T allowances, of $2.62. This is less than the overall average price-per-lb of the F-4370, as one would expect. The estimate does not directly address any incremental cost and weight of the rubber mounting isolators since that was judged to be virtually negligible and offset by minor modifications that decreased the weight of the original mounting brackets.

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*BBN designed the Freightliner two-stage mounts.

6.4 Exhaust System Costs

The baseline configuration of the F-4370 included the standard vertical, single aluminized muffler exhaust system. BBN replaced that system with a dual vertical muffler system, the components of which are described in Sec. 3. In this section we present the estimated price of the BBN modifications.

Table 10 lists the component modifications made by BBN to the original exhaust system. Note that the mufflers added by BBN are considerably heavier than the standard muffler. The mufflers mount on brackets affixed to the rear of the cab instead of masts mounted on the frame rail. This mounting system results in a smaller weight increase than typical mast mountings, which weigh approximately 40 lb [3,4].

While the F-4370 was delivered to BBN with the standard exhaust system, optional exhaust systems are available for it: A single vertical muffler "brite-finish" exhaust system, and a dual

TABLE 10. SUMMARY OF EXHAUST SYSTEM COMPONENTS AND WEIGHTS.

Component	Weight (lb)
Installed	
• Super Stack Silencers (2)	20.0
• Mufflers (2)	122.5
• Heat Shield & Bracket (1)	13.0
• Tee Can (1)	19.0
• 4 in. Piping (100 in.)	23.0
• 4 in. Flex Pipe (12 in.)	3.1
• Band Clamps	10.6
• Mounting Bracket	10.0
Removed	
• Original Muffler & Shield	<45.7>
• Exhaust Stack	<14.2>
• Exhaust Piping	<25.2>
Net Increase	136.1

vertical muffler "brite-finish" exhaust system. These IH options are summarized in Table 11. We used these systems as benchmarks for developing cost estimates for the BBN system.

There were two major adjustments to be made in order to use the IH optional exhaust systems as benchmarks. First, information on the current* prices of these options had to be converted to 1979 prices. Second, the IH options are for 5-in. systems, whereas BBN installed a 4-in. system. Therefore, the price of the 5-in. IH option has to be converted to a hypothetical 4-in. system. We then examined the differences between the BBN and

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*January 1981 price lists.

TABLE 11. COMPARISON OF AVAILABLE EXHAUST SYSTEM OPTIONS.

Type of Exhuast System	Additional Weight (lb)	List Price Increase ($)
Single Vertical Exhaust System • Aluminized Muffler, Tailpipe and Guard	-	-
Single Vertical 5-in. Exhaust System • Brite Finish (Option 07533)	13	260*
Dual Vertical 5-in. Exhaust System • Brite Finish (Option 07066)	126	568*
BBN Dual 4-in. Vertical Exhaust System	151	543

*Estimated 1979 prices based on January 1981 list prices ($310 and $677) and Producer Price Index for motor vehicles and equipment: 1979 Annual and December 1980.

IH 4-in. dual exhaust systems (e.g., Super Stack Silencers versus tail pipes) and estimated the net cost difference on the basis of the cost of components added and deleted.

The first adjustment was made by applying the Producer Price Index for motor vehicles and equipment to the January 1981 prices published by IH to restate the prices in 1979 dollars. The 1981 price for a brite-finish single exhaust system was reduced from $310 to $260; the brite-finish dual exhaust system was reduced from $677 to $568.

The conversion of the IH 5-in. systems to 4-in. system was based on differences in prices of 4-in. and 5-in. components. Donaldson had supplied to BBN confidential price information to be used "for computational purposes." Prices for a variety of 4-

in. and 5-in. components were included in the Donaldson price list. BBN calculated the ratio of the price of a 4-in. component to the price of a 5-in. component for a variety of components. The ratios ranged from 0.52 to 0.85, clearly a wide variation. Mufflers, the largest single cost of an exhaust system, were excluded from the analysis since they are easily adapted to either a 4-in. or 5-in. system. To be on the conservative side, i.e., not to underestimate the cost of the BBN system, 0.85 was taken as the factor to reduce the costs of the 5-in. system to a 4-in. system. The estimated prices in 1979 for hypothetical 4-in. brite-finish single and dual exhaust systems were estimated to be $221 and $482 respectively.* The latter figure was taken as a benchmark upon which to estimate the price of the BBN system.

Two basic changes were made to the optional IH brite-finish exhaust system that could affect the price. First, the "wye" pipe connection that splits the exhaust into two pipes was replaced by a "Splitter Tee Can." Second, Super Stack Silencers were installed on the mufflers in place of straight exhaust pipes. The mufflers, heat shields, and other components installed by BBN are essentially the same as would be found on the IH optional dual exhaust system. BBN also added more seal clamps than would be found on the IH dual system.

BBN estimated the net cost difference of the IH and BBN dual exhaust systems using the supplier price data that had been supplied by Donaldson. We assume a price markup of 1.4 at the manufacturer level and 1.35 at the dealer level, following the procedure used previously [3,4]. The net manufacturer price increase of the BBN brite-finish dual exhaust system over the comparable IH system is estimated to be $61. Given the estimated 1979 price of a 4-in. dual vertical exhaust system, $482, BBN estimates that the BBN system would carry a $543 dealer price over a standard exhaust system.

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*$260 x 0.85 = $221; $568 x 0.85 = $482.

REFERENCES

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3. E.K. Bender, J.W. Ernest, and J.A. Kane, "Noise Reduction Technology and Costs for a Ford CLT 9000 Heavy-Duty Diesel Truck," Bolt Beranek and Newman Inc. Report No. 4379, August 1980.

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5. "Buses: Noise Emission Standards for Transportation Equipment," Federal Register, Vol. 42, No. 176, pp. 45776-45797,12 September 1977.

6. "40 CFR 205: Transportation Equipment Noise Emission Controls," Federal Register, Vol. 41, No. 72, 13 April 1976.

7. Society of Automotive Engineers Recommended Practice SAE J336a, "Sound Level for Truck Cab Interior."

8. G.E. Fax and M.C. Kaye "The Economics of Quieting the Freightliner Cab-Over-Engine Diesel Truck," U.S. Department of Transportation Report No. DOT-TST-75-22, October 1974.

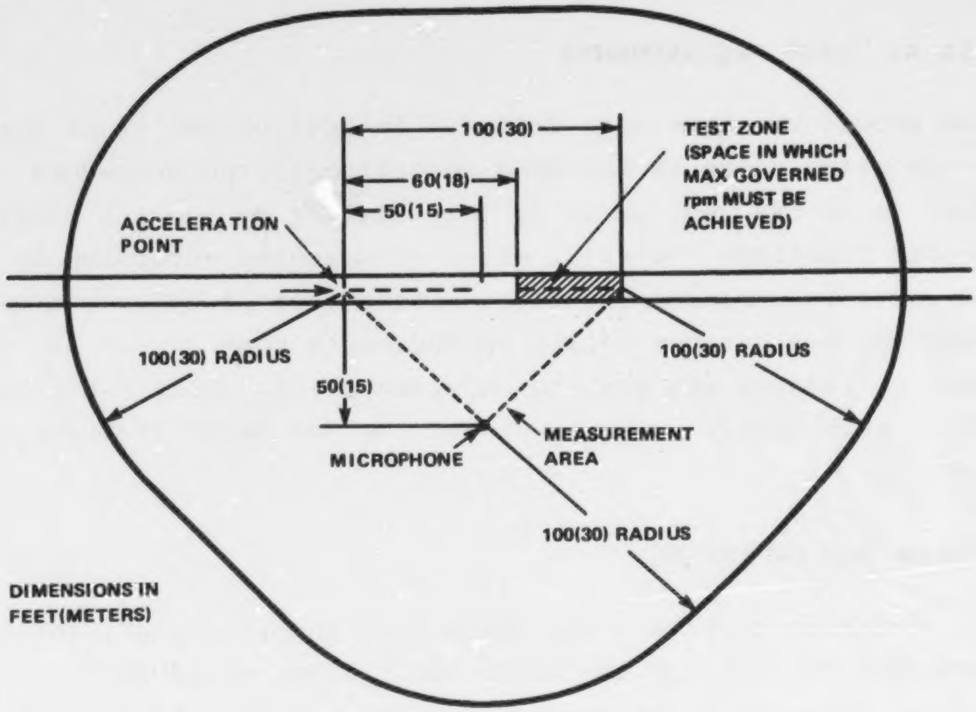
9. "Background Document for Medium and Heavy Truck Noise Regulations," U.S. Environmental Protection Agency, Report EPA-550/9-76-008, March 1976.

APPENDIX A: TEST REQUIREMENTS

Two procedures have been followed in testing the truck for noise. Exterior noise is measured according to the procedure described in 40 CFR 205, which is very similar to the SAE J366b Recommended Practice. Interior noise is measured according to the SAE J336a Recommended Practice. These test procedures are described in considerable detail in documents that should be consulted by readers who wish to understand them fully (see Refs. 5 and 8 of main report). Here we describe the major features of each test.

A.1 Exterior Test (40 CFR 205)

The exterior test is a low speed full throttle acceleration test intended to characterize drive train noise while de-emphasizing tire and aerodynamic noise [6]. The general arrangement of the test site is illustrated in Fig. A.1. The site is comprised of a paved vehicle path and measurement area, surrounded by an area that is free of reflecting objects. A microphone is located 4 ft above the ground and 50 ft from the center of the vehicle path. During a test, the vehicle is driven along a straight path at a constant speed corresponding to approximately two-thirds of governed engine speed. At the Acceleration Point the throttle is opened fully. The vehicle accelerates through the next 100 ft, reaching maximum governed rpm in the test zone. The truck is operated in the highest gear step that will permit it to meet this requirement. The peak noise level is generally measured twice on each side and the highest of the average values for each side is reported. Precision sound measuring equipment is used to ensure that accurate data are acquired.


100(30)
60(18)
50(15)
TEST ZONE (SPACE IN WHICH MAX GOVERNED rpm MUST BE ACHIEVED)
ACCELERATION POINT
100(30) RADIUS
100(30) RADIUS
50(15)
MEASUREMENT AREA
MICROPHONE
100(30) RADIUS

DIMENSIONS IN FEET(METERS)

FIG. A.1. TEST SITE FOR EXTERIOR NOISE LEVEL MEASUREMENTS.

For the noise data reported here the following operating conditions apply:

Engine Speed	- approach:	1200*rpm
	- final:	2250 rpm
Vehicle Speed	- approach:	11 mph
	- final:	20 mph
Gear Speed:		5th*

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*The gear step and approach engine speed were determined experimentally as required by the test procedure. It was found that when the truck approached in sixth gear, with the engine running two-thirds of governed speed, the engine reached governed speed when the vehicle was beyond the test zone. In fourth gear and two-thirds of governed speed, the engine reached governed speed before the test zone. Accordingly, the engine speed at approach was successively reduced in 100 rpm increments until it was found that, at 1200 rpm, governed speed was reached within the test zone.

An important feature of this test procedure is that it allows thermostatically controlled radiator fans to remain inoperative. Accordingly, the fan clutch hub was disengaged. This permitted the fan to turn only at a low speed at which its noise contribution was judged inconsequential.

A.2 Interior Test (SAE J336a)

The SAE J336a Recommended Practice specifies noise measurements 6 in. from the driver's ear while the truck is accelerating at full throttle from approximately 25 mph to 50 mph [7]. The gear step is selected so that the engine reaches rated speed at 50 mph. The test is performed with windows and vents closed and accessories turned off. Because of the relatively high speed at which the test is conducted, one may expect tire noise to be a more significant part of the total measured level than in the case of the 40 CFR 205 or SAE J366b test procedures.

The SAE J336a test procedure does not require the reporting of the A-weighted level, but rather the average of the two highest levels in each octave frequency band. The following table illustrates the band center frequencies for which measurements are to be acquired and the band pressure levels to be considered during the development of new vehicles.

Octave Band Center Frequency (Hz)	Band Pressure Level (dB)	Octave Band Center Frequency (Hz)	Band Pressure Level (dB)
63	101.5	1000	79.5
125	96.0	2000	74.0
250	90.5	4000	70.0
500	85.0	8000	70.0

The Recommended Practice states that "Trucks meet the design criteria if the sum of reported band pressure levels does not exceed the sum of the criteria band pressure levels, provided that no reported band pressure level exceeds the corresponding criteria band level by more than 3 dB." While the Recommended Practice does not specify an A-weighted criterion, the (logarithmic) sum of the A-weighted values of the band pressure levels specified in the above table is 87.6 dBA.

APPENDIX B: ESTIMATION OF NOISE SOURCE CONTRIBUTIONS IN THE IH F-4370

In this appendix, we describe how the contributions from the noise sources on the IH F-4370 were estimated from the various field measurements that were carried out on the truck. The estimates here are for source strengths when the truck is operated according to the SAE J366b test procedure. Table B.1 presents a description of each source and the variables that will be used in what follows to represent each.

TABLE B.1. NOISE SOURCES ON THE IH F-4370.

Variable	Source Description
EX	Exhaust outlet and shell noise
I	Engine intake noise
CB	Coastby noise, i.e., tires and drive train
E/T	Airborne and structureborne engine and transmission noise
ENB	Airborne noise coming from the back opening of the enclosure
ENF	Airborne noise coming from the front opening of the enclosure
ENR	Residual airborne noise escaping from the enclosure after the front and rear are sealed
SBR_i	Structureborne noise from the engine and transmission passing through the rear engine mounts, i = 1; single-stage mounts, i = 2; two-stage mounts
SBO	Structureborne noise from the engine and transmission not passing through the rear engine mounts, i.e., passing through the front engine mounts, transmission brackets, drive shaft, etc.

B.1 Strengths Before Treatment

Before installation of the noise control treatments, the total noise from the truck, N', is given by*

$$N' = EX' \oplus I' \oplus CB' \oplus E/T' \tag{B.1}$$

where the prime refers to the source strength prior to installation of the noise control treatments and the variables are explained in Table B.1. Exhaust (EX') and intake (I') source strengths were estimated from laboratory data provided by the Donaldson Co. as described in the text of this report. Tire and drive train source strength (CB') were estimated from measurements of the noise from the truck as it coasted by a microphone 50 ft from the truck centerline with the engine off. When operating the truck according to the SAE J366b test procedure, we found that the maximum noise occurred when the truck was opposite the microphone. At that point in the test the truck was generally going about 20 mph. Consequently, the maximum noise during a truck coastby at 20 mph was used to estimate the tire/drive train source strength. Section 2 of the report presents that coastby data. Engine/transmission source strength (airborne and structureborne combined) was estimated by subtracting EX', I', and CB' from the overall truck noise measured according to the SAE J366b test procedure. Table B.2 presents the results of the above tests and calculations.

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*The symbol $\oplus$ refers to logarithmic addition defined by

$$A \oplus B = 10 \log \; 10^{A/10} + 10^{B/10}$$

TABLE B.2. NOISE SOURCE STRENGTHS ON IH F-4370 BEFORE TREATMENT.

Source Description	Source Strength (dBA)	Source Variable
Exhaust	74	EX'
Intake	47	I'
Tires/Drive Train	60	CB'
Engine	80.1	E/T'
Total	81.1	N'

B.2 Source Strengths After Treatment

After installation of the noise control treatments, the total noise from the truck is given by

$$N = EX \oplus I' \oplus CB' \oplus ENF \oplus ENB \oplus ENR \oplus SBR2 \oplus SBO \qquad (B.2)$$

where it has been assumed that intake, tire, and drive train noise are unaffected by the improved exhaust system, engine enclosure, and two-stage rear engine mounts that constitute the noise control treatments.

B.3 Structureborne Noise

For this calculation we have divided the structureborne noise from the engine and transmission into two parts. The structureborne sound that passes through the rear engine mounts (SBR) has been separated from the structureborne sound passing through all other paths (SBO), i.e., the drive shaft, front engine mount, and the transmission bracket.* To help quantify the structureborne contribution of the engine and transmission, we carried out a series of tests in which we jacked up the engine off its front mount, disconnected the drive shaft, and removed the transmission bracket. We began by first measuring the noise from the truck in its baseline condition, but with the front and rear of the enclosure sealed with leaded vinyl and fiberglass. Two microphones were located 50 ft from the truck centerline opposite the exhaust stack on each side of the truck. Accelerometers were placed at various critical locations on the frame rails. At high idle the noise level on the left side of the truck was found to be 72 dBA. After disconnecting the transmission bracket and drive shaft and jacking up the engine off its front mount, we measured 71.1 dBA. The residual structureborne noise can be found approximately from the differences in these two measurements,† i.e.,

$$\text{SBO} = 72 \ominus 71.1 = 64.7 \text{ dBA} \quad . \qquad (\text{B.3})$$

Of course the above result is strictly correct only for the engine operating at high idle. Fortunately, the vibration on the frame rail of the truck as illustrated in Fig. B.1 is very nearly the same whether one measures the vibration with the truck operating at high idle, or whether one measures the vibration just as the sound level peaks while operating the truck according

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*The transmission bracket is a steel bar with each end resting on a shelf welded to the frame rail. The middle of the bar is bolted to the transmission. Its purpose is to restrain the transmission while operating under load and thereby prevent the transmission from jumping out of gear.

†The symbol $\ominus$ means logarithmic subtraction defined by

$$A \ominus B = 10 \log \ 10^{A/10} - 10^{B/10}$$

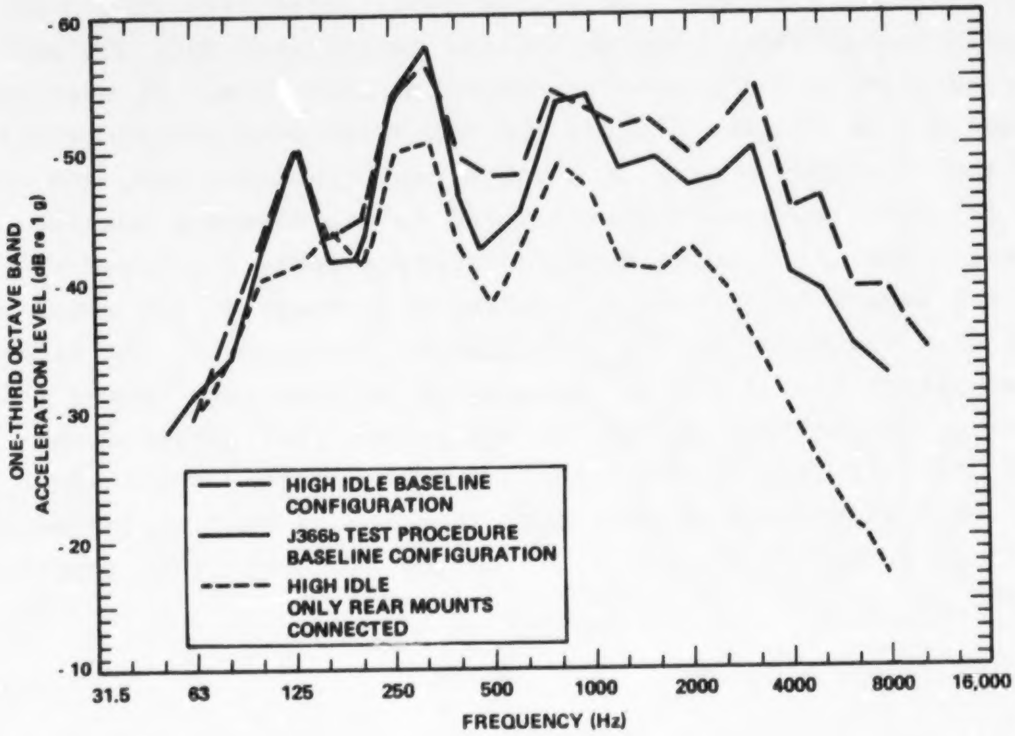

FIG. B.1. VIBRATION LEVEL ON FRAME RAIL.

to the SAE J366b test procedure. Also shown in the figure is the frame rail vibration after jacking up the engine, disconnecting the drive shaft and removing the transmission bracket. The vibration is reduced considerably, indicating that these three paths do significantly influence the structureborne sound from the truck. However, the small change in the noise level when these three paths were disconnected indicates that structureborne sound is not as important as other sources in generating the overall truck noise.

To estimate the transmission of structureborne sound through the rear mounts we carried out a series of measurements with the truck equipped with its improved exhaust system (see Sec. 3 of the main text), two-stage rear engine mounts (see Sec. 3), and an early version of the engine/transmission enclosure. We measured the noise from the truck using the SAE J366b test procedure first with the two-stage mounts in their normal configuration, and then with the blocking mass shortcircuited to the flywheel housing bracket. The latter configuration approximately simulated the original single stage mounts. Taking an average of ten runs, we found that the noise on the left side of the truck in the first configuration was 74.7 dBA. An average of five runs showed that the noise similarly measured for the second configuration was 75.08 dBA. If SBR_1 is the structureborne source strength for vibration transmitted through the single stage rear engine mounts and SBR_2 is that source strength for the two stage rear mounts, then we can write

$$SBR_1 \ominus SBR_2 = 75.08 - 74.7 = 64.3 \quad . \qquad \text{(B.4)}$$

If ΔSBR is the insertion loss of the two-stage mount such that

$$SBR_1 - SBR_2 = \Delta SBR \qquad \text{(B.5)}$$

then, after some simple algebraic manipulations, we can write

$$SBR_1 = 64.3 - 10 \log (1 - 10 -\Delta SBR/10) \qquad \text{(B.6)}$$

and

$$SBR_2 = SBR_1 - \Delta SBR \quad . \qquad \text{(B.7)}$$

To estimate ΔSBR, the reduction in structureborne sound through the rear engine mounts due to the use of the two-stage isolator, we have used the analytical estimate of the one-third octave band spectrum of structureborne noise from the engine and transmission obtained as described in Appendix C. That estimate is strictly correct only for the sum of the structureborne noise from all paths. Also, it is in effect an upper bound estimate of the structureborne noise since the radiation efficiencies are all assumed to be one. Nevertheless, if we are willing to assume that the spectral content of the structureborne noise passing through the single-stage rear mount is similar to the analytical estimate in Appendix C, then we can estimate ΔSBR by using the one-third octave band insertion loss data for the two-stage mount in Fig. 25 of the text. In Fig. B.2 we show the result of

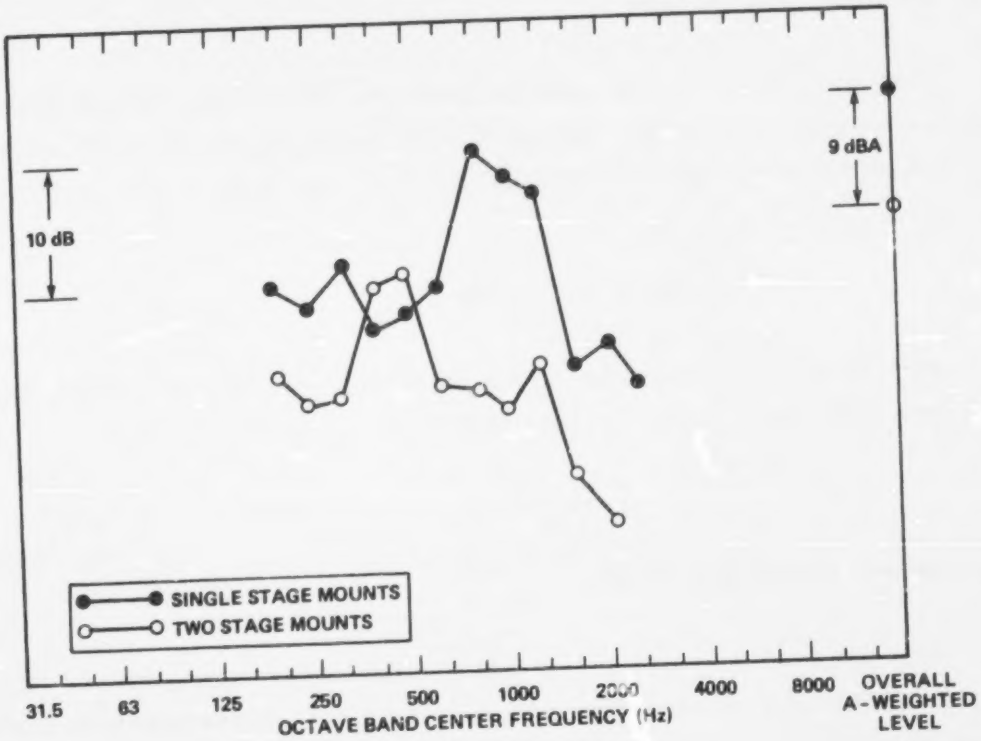

FIG. B.2. ANALYTICAL ESTIMATE OF THE CHANGE WITH A-WEIGHTED ONE-THIRD OCTAVE BAND.

subtracting that insertion loss from the analytical estimate of the structureborne noise spectrum. The result is a 9-dBA reduction in structureborne noise passing through the rear mount, i.e.,

$$\Delta SBR = 9 \text{ dBA} \quad .$$

Using the result in Eqs. B.6 and B.7, we obtain

$$SBR_1 = 64.8$$

$$SBR_2 = 55.8 \quad .$$

B.5 Exhaust and Intake

The strength of the exhaust outlet and shell noise from the improved exhaust system has been estimated based on laboratory measurements made by the Donaldson Co., as described in Sec. 3 of the text, i.e.,

$$EX = 59.5 \text{ dBA} \quad .$$

The intake system was unchanged and, hence, the same source strength was used here as for the untreated truck, i.e.,

$$I = 47 \text{ dBA} \quad .$$

B.6 Drive Train and Tires

The noise from the drive train and tires was assumed to remain unchanged after treating the truck. Consequently, based on coastby data we have

$$CB = 60 \text{ dBA} \quad .$$

B.7 Engine/Transmission Airborne Noise

The enclosure is a flow-through design extending from the radiator to the back of the cab. Large openings are provided in the front and rear to admit cooling air. The airborne noise from the engine and transmission is divided into the noise that comes through the front opening of the enclosure, ENF; the noise that comes through the back opening of the enclosure, ENB; and the residual noise, ENR. The last is the noise that is transmitted through the walls of the enclosure, or through leaks in the walls of the enclosure due to imperfect seals between adjacent panels, or through holes where components such as hoses pass through the panels. To estimate the strength of these three sources, we made a series of noise measurements with the truck operating according to the SAE J366b test procedure and with the front and rear openings alternatively open and sealed with leaded vinyl. Table B.3 shows the results of these measurements for the left side of the truck, the noisy side. The noise levels in the table are the result of averaging two to four runs for each condition. If we assume that sealing the front and rear openings totally eliminates the noise from those paths, we can readily calculate ENF and ENB from the data in Table B.3 in two different ways. For example,

$$\text{ENF} = (\text{Noise with Back and Front Open}) - (\text{Noise With Back Open and Front Closed}) \tag{B.8}$$

and alternatively

$$\text{ENF} = (\text{Noise with Back Closed and Front Open}) - (\text{Noise with Front and Back Closed}) \;. \tag{B.9}$$

TABLE B.3. TRUCK NOISE LEVELS FOR ALTERNATIVE ENCLOSURE CONFIGURATIONS.

Condition	Average Left Side Noise Levels (dBA)	No. of Runs
Back and Front Closed	74.0	3
Back Closed/Front Open	74.5	3
Back Open/Front Closed	74.6	4
Back Open/Front Open	75.1	2

Similar calculations can be made for ENB. Table B.4 presents the estimates of ENF and ENB using the above two equations and two similar equations for ENB. Since the two result in slightly different source strength values, the table also shows the average of the two values. The residual airborne noise from the enclosure ENR can now be calculated from Eq. B.2 by using the previously calculated values of the various source strengths and the measured overall noise from the fully treated truck, i.e.,

$$N = 72.7 \text{ dBA}$$

and

$$\text{ENR} = 72.7 \ominus \text{Ex} \ominus \text{I} \ominus \text{CB}' \ominus \text{ENF} \ominus \text{ENB} \ominus \text{SBR}_2 \ominus \text{SBO} \quad .$$

The calculated values of ENR can be found in Table B.4. Table B.5 summarizes all the source strengths for the treated truck. In Table B.6, we have combined these source strengths so as to compare the truck in three configurations: untreated, treated but with single-stage mounts, and fully treated.

TABLE B.4. ENGINE/TRANSMISSION AIRBORNE SOURCE STRENGTHS.

Source	Maximum Source Strength (dBA)	Minimum Source Strength (dBA)	Average Source (dBA)
ENB	66.2	65.7	66.0
ENF	65.5	64.9	65.2
ENR	67.7	68.3	68.0

TABLE B.5. SUMMARY OF SOURCE STRENGTHS OF TREATED IH F-4370.

Source	Variable	Source Strengths (dBA)
Exhaust	EX	59.5
Intake	I	47
Tires and Drive Train (coastby)	CB	60
Airborne noise from the back of the enclosure	ENB	66
Airborne noise from the front of the enclosure	ENF	65.2
Residual airborne noise from the enclosure	ENR	68
Structureborne noise through the single-stage rear engine mounts	SBR_1	64.8
Structureborne noise through the two-stage rear engine mounts	SBR_2	55.8
Residual structureborne noise	SBO	64.7

TABLE B.6. SOURCE STRENGTHS FOR THREE CONFIGURATIONS OF IH F-4370.

Sources	Baseline (dBA)	Treated Truck Single-Stage Mounts (dBA)	Treated Truck Two-Stage Mounts (dBA)
Exhaust	74.0	59.5	59.5
Intake	47.0	47.0	47.0
Tires and Drive Train	60.0	60.0	60.0
Engine/Transmission Airborne	-	71.3	71.3
Engine/Transmission Structureborne	-	67.8	65.2
Total Engine/Transmission	80.1	72.9	72.3
Overall Noise	81.1	73.3	72.7

APPENDIX C: PRELIMINARY ESTIMATION OF THE STRUCTUREBORNE NOISE FROM THE IH F-4370

Early in the program we were concerned that the structureborne noise from the engine and transmission might be a significant source of noise in the IH F-4370. To make a preliminary assessment of that source strength, we performed a vibration survey on the truck measuring the one-third octave band acceleration spectrum, AL(ω), at ten locations while operating the truck according to the SAE J366b test procedure. The locations measured were as follows:

- Bumper
- Fuel tank
- Battery box
- Cab
- Frame rail (six positions).

The measured spectra are shown in Figs. C.1 through C.3.

Taking the radiation efficiency of all these surfaces as unity, we can estimate the sound pressure level, SLP(ω), at frequency ω at 50 ft to be

$$\mathrm{SPL}(\omega) = 124 + \mathrm{AL}(\omega) + 10 \log A - 20 \log \omega \quad . \tag{C.1}$$

where A is the area of each radiating surface in square feet. The surface areas of the above elements are given in Table C.1. Using the acceleration levels in Figs. C.1 through C.3 and the areas in Table C.1 in Eq. C.1, we have estimated the structureborne noise in one-third octave frequency bands radiated by the truck. Figure C.4 presents that estimate and compares it to measurements of the noise radiated by the truck in the baseline configuration before installation of the enclosure or the two-stage engine mounts, but after installation of the improved

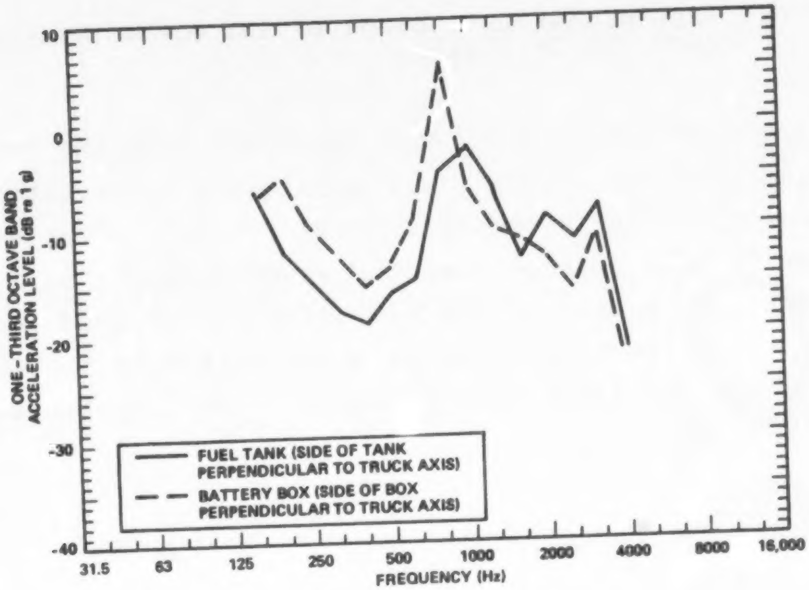

FIG. C.1. ONE-THIRD OCTAVE BAND ACCELERATION LEVELS ON FUEL TANK AND BATTERY BOX.

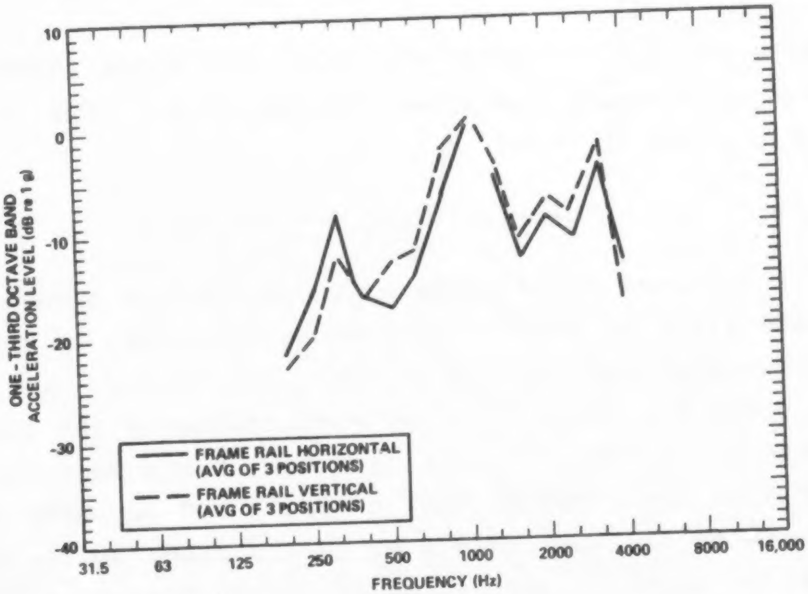

FIG. C.2. ONE-THIRD OCTAVE BAND ACCELERATION LEVELS ON FRAME RAIL.

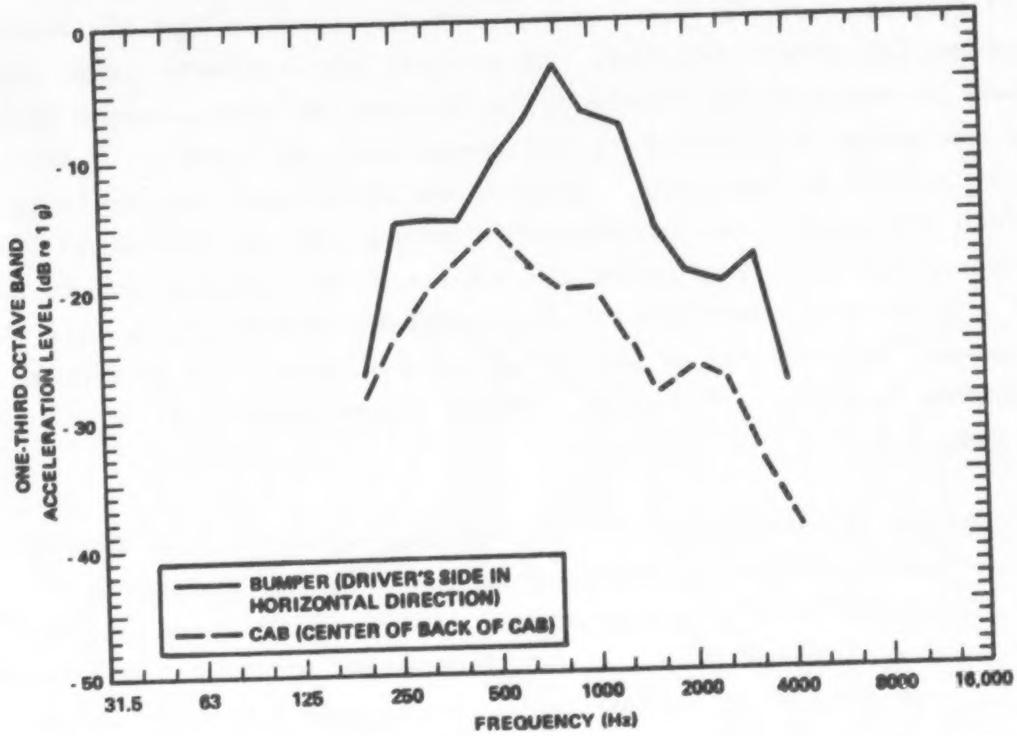

FIG. C.3. ONE-THIRD OCTAVE BAND ACCELERATION LEVELS ON BUMPER AND CAB.

TABLE C.1. SURFACE AREA OF TRUCK COMPONENTS.

	(ft2)
Bumper	14.5
Fuel Tanks	34.0
Battery Boxes	15.7
Cab (excluding doors)	21
Frame Rail (vertical)	23.7
Frame Rail (horizontal)	39.5

exhaust system (dual exhausts). The estimated structureborne noise, which is quite high, led us to believe initially that we had a more severe problem than we actually did. Figure C.4 shows that significant structureborne contributions in the low and mid frequencies were predicted. Of course, our estimate is an upper bound on the structureborne noise because we have assumed that the radiation efficiency of all components is unity. In fact, after modifying the truck further, we found that our estimate was indeed too high. We constructed a mockup of the engine enclosure using masonite to determine the acoustic performance and to bring out any special clearance or fit problems before going to a full metal enclosure. Noise measurements were made with the truck equipped with that enclosure. Those measurements are also shown in Fig. C.4.

Since the enclosure should not have affected the structureborne noise radiated by the components investigated in this appendix, the fact that our structureborne noise predictions exceed the measured noise levels in the 800 to 1000 and 1250 Hz bands indicates that our estimate is too high in those bands. Also, as we have shown in Appendix B, the structureborne noise based on other measurements appears to be about 67.8 dBA rather than the 73.7 dBA estimated here. On the other hand, it is encouraging to note that if we assume that the noise from the truck with the masonite enclosure is dominated by structureborne noise in the 800, 1000, and 1250 Hz bands and that our structureborne noise estimate is correct in all other frequency bands, then the overall structureborne noise that we predict is 67.2 dBA. That compares favorably with the 67.8 dBA estimate in Appendix B.

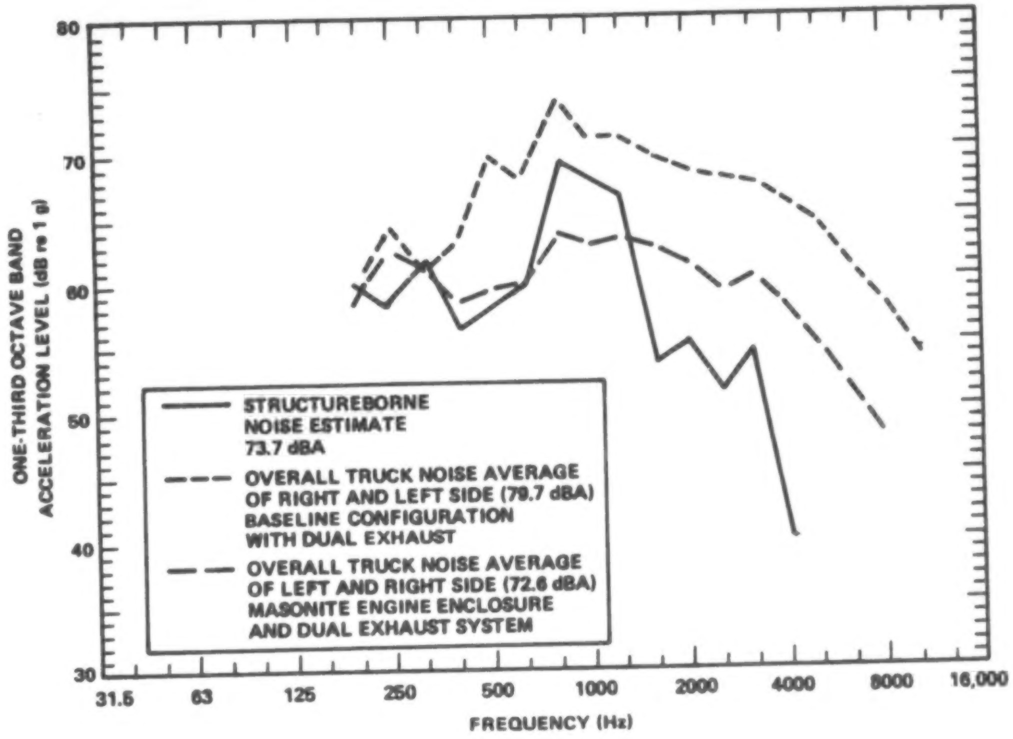

FIG. C.4. STRUCTUREBORNE NOISE ESTIMATE.